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ENERGY CONVERSION ALTERNATIVES STUDY
-ECAS-
GENERAL ELECTRIC PHASE I FINAL REPORT

VOLUME III, ENERGY CONVERSION SUBSYSTEMS AND COMPONENTS
Part 1, Bottoming Cycles and Materials of Construction

by

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16. Abstract

A parametric study was performed to assist in the development of a data base for the comparison of advanced energy conversion systems for utility applications using coal or coal-derived fuels. Estimates of power plant performance (efficiency), capital cost, cost of electricity, natural resource requirements, and environmental intrusion characteristics were made for ten advanced conversion systems. Over 300 parametric points were analyzed to estimate the potential of these systems. Emphasis of the study was on the energy conversion system in the context of a base loaded utility power plant. Although cases employing transported coal-derived fuels were included in the study, the fuel processing step of converting coal to clean fuels was not investigated except for cases where a low-Btu gasifier was integrated with the power plant. All power plant concepts were premised on meeting emission standards requirements. The investigative approach focused on achieving consistency and comparability in the analysis of the various conversion systems. Recognized advocate organizations were employed to analyze their respective cycles and to present their analyses for power plant integration by the GE systems evaluation team. Wherever possible, common subsystems and components for the various systems were treated on a uniform basis. A steam power plant (3500 psig, 1000 F, 1000 F) with a conventional coal-burning furnace-boiler was analyzed as a basis for comparison. Combined cycle gas/steam turbine system results indicated competitive efficiency and a lower cost of electricity compared to the reference steam plant. The Open-Cycle MHD system results indicated the potential for significantly higher efficiency than the reference steam plant but with a higher cost of electricity. The information contained in this report constitutes results from the first phase of a two phase effort. In Phase II, a limited number of concepts will be investigated in more detail through preparation of conceptual designs and an implementation assessment including preparation of R&D plans estimating the resources and time required to bring the systems to commercial fruition.

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FOREWORD

The work described in this report is a part of the Energy Conversion Alternatives Study (ECAS)—a cooperative effort of the Energy Research and Development Administration, the National Science Foundation, and the National Aeronautics and Space Administration.

This General Electric contractor report for ECAS Phase I is contained in three volumes:

Volume I - Executive Summary

Volume II - Advanced Energy Conversion Systems

- Part 1 - Open-Cycle Gas Turbines
- Part 2 - Closed Turbine Cycles
- Part 3 - Direct Energy Conversion Cycles

Volume III - Energy Conversion and Subsystems and Components

- Part 1 - Bottoming Cycles and Materials of Construction
- Part 2 - Primary Heat Input Systems and Heat Exchangers
- Part 3 - Gasification, Process Fuels, and Balance of Plant

In addition to the principal authors listed, members of the technical staffs of the following subcontractor organizations developed information for the Phase I data base:

General Electric Company
- Advanced Energy Programs/Space Systems Department
- Direct Energy Conversion Programs
- Electric Utility Systems Engineering Department
- Gas Turbine Division
- Large Steam Turbine-Generator Department
- Medium Steam Turbine Department
- Projects Engineering Operation/I&SE Engineering Operation
- Space Sciences Laboratory

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Foster Wheeler Energy Corporation

Thermo Electron Corporation

This General Electric contractor report is one of a series of three reports discussing ECAS Phase I results. The other two reports are the following: Energy Conversion Alternatives Study (ECAS), Westinghouse Phase I Final Report (NASA CR-134941), and NASA Report (NASA TMX-71855).

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Summary

ENERGY CONVERSION SUBSYSTEMS AND COMPONENTS

The objective of Phase I of the Energy Conversion Alternatives Study (ECAS) for coal or coal-derived fuels was to develop a technical-economic information base on the ten energy conversion systems specified for investigation. Over 300 parametric variations were studied in an attempt to identify system and cycle conditions which indicate the best potential of the energy conversion concept. This information base provided a foundation for selection of energy conversion systems for more in-depth investigation in the conceptual design portion of the ECAS study. The systems for continued study were specified by the ECAS Interagency Steering Committee.

The major emphasis of this study was the evaluation of the prime cycle portion of the energy conversion system. The energy conversion subsystems and auxiliary systems are coupled to the prime cycle to produce a complete power plant. These subsystems were applied to each of the prime cycles on a consistent basis. Each of the subsystems, e.g., furnaces, bottoming cycles, balance of plant, was analyzed by its respective independent study team for each specific application to an energy conversion system.

The furnace systems included both direct combustion of coal and combustion of process fuels derived from coal. The furnaces with direct coal combustion employing fluidized beds with in-bed sulfur capture appear to be the most attractive options for the closed-cycle advanced energy conversion systems.

Both organic and steam cycles were studied for bottoming many of the prime cycles. The characteristics of the organic cycles made them most attractive in ratings up to 100 MWe and peak organic cycle temperature less than 500 F (533 K). Although the addition of an organic bottoming cycle to a prime cycle showed an efficiency improvement, a relatively high capital cost addition for the organic bottoming cycle and its related balance of plant was estimated. A steam bottoming cycle was an essential requirement for use with many of the prime cycles; e.g., Combined Cycle Gas Turbine, Liquid Metal Topping Cycle, MHD Systems, and High-Temperature Fuel Cells. The steam bottoming cycles were all analyzed by the same study team to assure a uniform assessment. Steam throttle conditions and feedwater heating chains were varied, however, to accommodate specific prime cycle requirements for improvement of the system efficiency.

In energy conversion systems which could utilize coal directly, the employment of clean fuels produced from coal did not appear to be economically attractive. In systems which require a fuel processing step, e.g., open-cycle gas turbines, the semi-clean liquid fuels produced from coal appeared to be an attractive alternative and were close to an economic standoff with the low-Btu integrated gasifier technique for producing an acceptable gas turbine fuel.

Introduction

ENERGY CONVERSION SUBSYSTEMS AND COMPONENTS

Many advanced energy conversion techniques which can use coal or coal-derived fuels have been advocated for power generation applications. Conversion systems advocated have included open- and closed-cycle gas turbine systems (including combined gas turbine-steam turbine systems), supercritical CO_2 cycle, liquid metal Rankine topping cycles, magnetohydrodynamics (MHD), and fuel cells. Advances have also been proposed for the steam systems which now form the backbone of our electric power industry. These advances include the use of new furnace concepts and higher steam turbine inlet temperatures and pressures. Integration of a power conversion system with a coal processing plant producing a clean low-Btu gas for use in the power plant is still another approach advocated for energy conserving, economical production of electric power. Studies of all these energy conversion techniques have been performed in the past. However, new studies performed on a common basis and in light of new national goals and current conditions are required to permit an assessment of the relative merits of these techniques and potential benefits to the nation.

The purpose of this contract is to assist in the development of an information base necessary for an assessment of various advanced energy conversion systems and for definition of the research and development required to bring these systems to fruition. Estimates of the performance, economics, natural resource requirements and environmental intrusion characteristics of these systems are being made on as comparable and consistent a basis as possible leading to an assessment of the commercial acceptability of the conversion systems and the research and development required to bring the systems to commercial reality. This is being accomplished in the following tasks:

Task I	Parametric Analysis	(Phase I)
Task II	Conceptual Designs	(Phase II)
Task III	Implementation Assessment	

This investigation is being conducted under the Energy Conversion Alternatives Study (ECAS) under the sponsorship of Energy Research and Development Administration (ERDA), National Science Foundation (NSF), and National Aeronautics and Space Administration (NASA). The control of the program is under the direction of an Interagency Steering Committee with participation of the supporting agencies. The NASA Lewis Research Center is responsible for project management of this study.

The information presented in this report describes the results produced in the Task I portion of this study. The emphasis in this task was placed upon developing an information base upon which comparisons of Advanced Energy Conversion Techniques using coal or coal-derived fuels can be made. The Task I portion of the study was directed at a parametric variation of the ten advanced energy conversion systems under investigation. The wide-ranging parametric study was performed in order to provide data for selection by the Interagency Steering Committee of the systems and specific configurations most appropriate for Task II and III studies.

The Task II effort will involve a more detailed evaluation of seven advanced energy conversion systems and result in a conceptual design of the major components and power plant layout. The Task III effort will produce the research and development plans which would be necessary to bring each of the seven Task II systems to a state of commercial reality and then to assess their potential for commercial acceptability.

A prime objective of this study was to produce results which had a cycle-to-cycle consistency. In order to accomplish this objective and still ensure that each system was properly advocated, an organization which is or had been a proponent of the prime cycle was selected to advocate the energy conversion system and to analyze the performance and economics of the prime cycle portion of the energy conversion system, i.e., the parts of the system which were novel or unique to the system. The remaining subsystems, e.g., fuel processing, furnaces, bottoming cycles, balance of plant, were analyzed by technology specialist organizations which presently have responsibility for supplying these subsystems for utility applications. The final plant configuration and performance were produced by the General Electric Corporate Research and Development study team and this group performed the critical integration of the final plant concept. This methodology was used to provide a system-to-system consistency while maintaining the influence of a cycle advocate.

The energy conversion subsystems and components which were applied on a common basis to each of the advanced energy conversion systems are described in this Volume. The discussion and results for each of the advanced systems is given in Volume II.

Bottoming Cycles are applied to most of the advanced energy conversion systems. To the maximum extent possible, the bottoming cycles were assumed to be composed of state-of-the-art components. Steam bottoming cycles are utilized for "high-temperature" applications bottoming with steam conditions being limited to 1000 F (811 K). Organic fluid bottoming cycles are employed for the low-temperature applications (temperatures less than 600 F [589 K]).

The Materials of Construction are defined for each of the energy conversion systems. This includes both the identification of the materials and the assumptions which were made with respect to design criteria.

Primary Heat Input Systems were employed for all closed-cycle applications. The heat exchanger equipment provides for the transport process to introduce thermal energy into the cycle working fluids. Advanced furnace techniques for direct combustion of coal and combustion of clean fuels were considered. The atmospheric fluidized bed with direct coal was utilized as a reference furnace for the closed-cycle parametric variations.

Heat Exchangers were employed in all advanced energy conversion systems. This fluid-to-fluid exchange equipment provided for transport processes within the cycles, e.g., the regeneration of thermal energy, heat rejection precoolers, and low temperature air preheaters.

Gasification and Process Fuels derived from coal were employed as clean fuel sources for combustion systems. The low-Btu gasifier employed for integrated plants was the fixed bed gasifier with low-temperature cleanup. The process fuels were considered as delivered to the plant boundary. The cost and conversion efficiency for these clean fuel production processes were directly related to the fixed bed gasifier. This gave a basis for cost comparison between the use of process fuels and integrated gasifier systems.

The Balance of Plant for the advanced energy conversion concepts considered the installation of the specific components of the energy conversion cycle and primary heat input heat exchangers and the supply and installation of the auxiliary plant equipment. The fuel supply and storage system and the heat rejection system were two of the major elements evaluated as balance-of-plant items.

Section 4

BOTTOMING CYCLES

INTRODUCTION

A bottoming cycle operates in conjunction with a prime cycle to extract useful work from the otherwise waste energy of a prime thermodynamic cycle. In contrast with a prime cycle, the selection of a bottoming cycle, and its operating parameters for maximizing the overall cycle efficiency, is guided less by the efficiency of the bottoming cycle than by the ability to maximize the total cycle output and efficiency.

Consider, for example, the closed-cycle gas turbine as a prime cycle (see Figure 4-1). The thermodynamic efficiency (η_p) of this cycle is:

$$\eta_p = \frac{P_1}{Q_1}$$

and the energy rejected $Qr_1 = (1 - \eta_p) \cdot Q_1$. Here, P_1 is the net energy output and Q_1 is the energy input for the prime cycle.

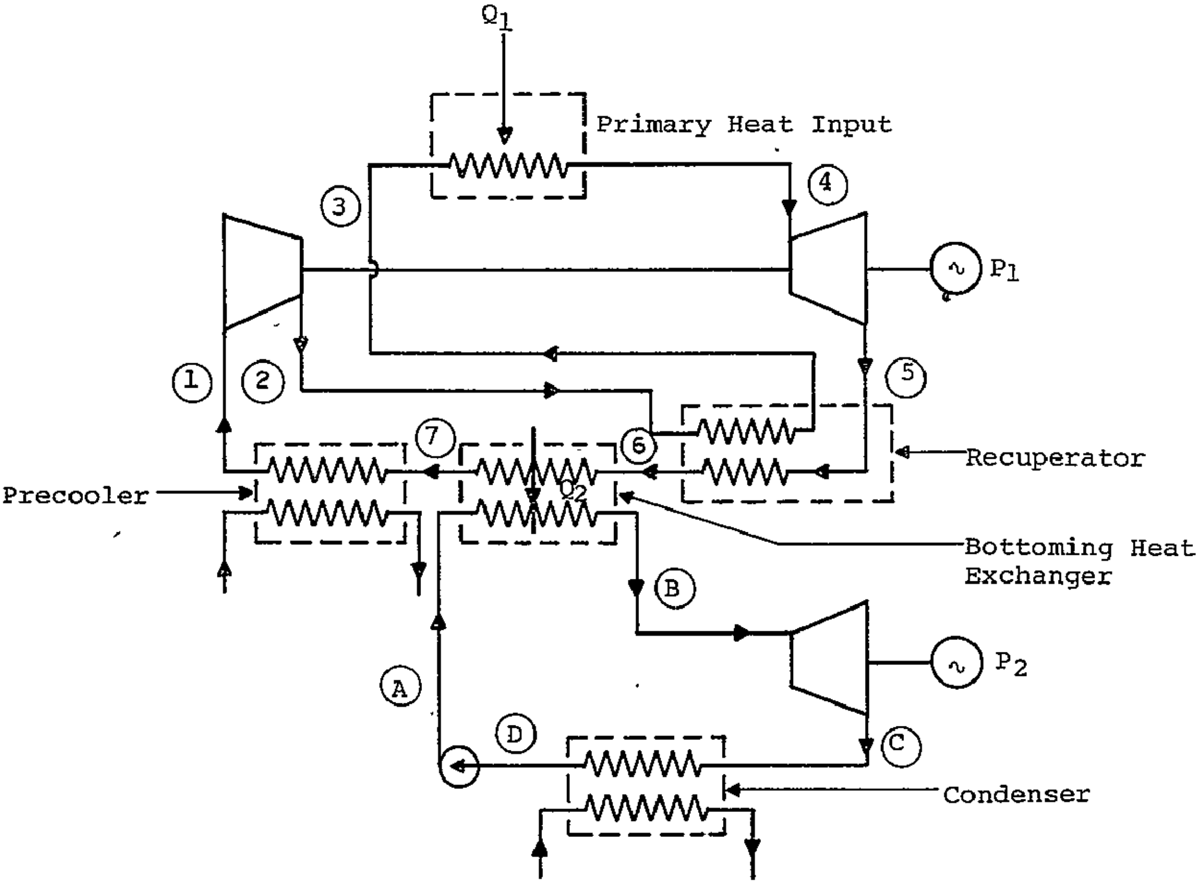

Figure 4-1. Typical Integration of Primary and Bottoming Cycles

The thermal energy, Qr_1, represents the energy in prime-cycle fluid between state points 6 and 1, which must be rejected for the prime cycle to conform to the second law of thermodynamics. If a bottoming cycle is employed, it will extract a fraction, f, of the waste energy Qr_1; hence, extracted energy $Q_2 = f \cdot Qr_1$.

If the efficiency of the bottoming cycle is η_s, the overall combined-cycle thermodynamic efficiency is:

$$\eta_{overall} = \eta_p + f\,(1 - \eta_p)\,\eta_s$$

This is based on negligible furnace losses, auxiliary losses, etc.

The importance of the above expression is clear from the following observations:

a. In cases where a bottoming cycle can be employed, it is not always necessary to utilize such means as regeneration to improve the prime-cycle efficiency if the bottoming cycle can successfully exploit the energy potential of the prime-cycle flow stream.

b. Inasmuch as the bottoming cycle efficiency, η_s, and the fraction, f, of the reject energy, Qr_1, improve the overall cycle efficiency in an identical manner, an optimum bottoming cycle would be one that maximizes ($f \cdot \eta_s$) for a given prime cycle. Thus,

   - Maximum energy fraction, f, is achieved by cooling the prime fluid to the lowest possible temperature level (at point 7 in Figure 4-1). Since the temperature of bottoming fluid at A is set by the cooling medium in the condenser, the temperature at 7 is primarily influenced by design limitations of the heat exchanger. This is discussed in detail later in this section under "Energy Extraction Limits—Pinch-Point Problem."
   - Since in most cases larger increases can be achieved in the energy fraction, f, than in the efficiency, η_s, the bottoming cycles are designed to be as simple as possible; for example, without substantial regeneration of the working fluid. This approach would also keep additional plant complexity to the minimum.

TYPES OF BOTTOMING CYCLES

The following bottoming cycles are considered for various prime energy conversion cycles studied in this program.

Large Steam Bottoming Cycle

For all systems in which a large, high-performance steam bottoming cycle could be effectively employed, the bottoming cycle used had turbine throttle steam conditions of 3500 psig (24.2×10^6 N/m^2) at 1000 F (811 K), with a reheat stage at 1000 F

(811 K). The supercritical steam cycle conditions are frequently those specified by utility customers. These conditions also result in a steam plant with very attractive efficiency (heat rates). Output capacity of 400 to 1100 MWe are typical of current steam plants. A lower limit of 250 MWe is selected for the purpose of this study.

Large steam bottoming cycles are selected for those prime cycles which have significant topping cycle reject heat and temperatures high enough to allow steam reheat in the bottoming cycle.

The choice of final feedwater temperature (FFWT) level for a bottoming cycle is influenced by the temperature-enthalpy characteristics of the prime-cycle reject fluid stream and by other economic considerations in the design of interface heat exchangers. Thus, the regenerative feedwater heating train used for a bottoming cycle may have significantly fewer than the six to eight feedwater heaters common to a modern central steam plant.

Medium Steam Bottoming Cycle

The steam conditions for these bottoming cycles are at the subcritical level (1250 to 2400 psig [8.7×10^6 to 16.6×10^6 N/m^2], 950 to 1000 F [783-811 K]), generally without provision for reheat. These cycles are applicable when the prime-cycle exhaust fluid temperature is in the range of 500 F to 1500 F (533 K to 1089 K), and a minimum primary fluid flow of about 0.5 million lb/hr (0.23 million kg/hr). Reheat is considered only when the prime fluid exhaust temperature is above 1100 F (866 K). A total bottoming cycle output of less than 250 MWe is typical for such steam plants.

Because of the subcritical pressure level operation, these steam cycles offer significant flexibility for integration with certain prime cycles (e.g., open-cycle gas turbine cycles—air and water cooled) which require a steam supply for their operation. This flexibility arises from the fact that a dual-pressure (dual-drum) steam boiler can be employed, which would supply saturated process steam from the low-pressure drum, and would have the potential to lower stack temperature concurrently.

Organic Bottoming Cycle

The organic bottoming cycles are most effectively employed when the prime fluid exhaust temperature is less than 900 F (755 K). With a judicious selection of an organic cycle working fluid, the primary cycle fluid stream can be cooled down to a low temperature level (dictated by factors external to the bottoming cycle), thus achieving a maximum energy recovery fraction, f. Since the thermodynamic efficiency of the organic cycle, η_s, is comparable to that of the steam bottoming cycle under similar conditions, the overall cycle efficiency improvement is significantly better with an organic bottoming plant than with a steam bottoming plant.

Organic cycles suffer from one major limitation. Because the chemical decomposition rate for most organic fluids becomes significant above 600 F to 700 F (589 K to 644 K), these cycles cannot utilize primary fluid at exhaust temperature levels much above this safe operating limit. Selection of a proper organic fluid is based on such considerations as appropriate thermodynamic characteristics, safety, and economics. Each candidate organic fluid has a certain prime-cycle fluid temperature range in which the organic fluid offers maximum energy extraction relative to other fluids in the same temperature range. Organic cycles are unique in that they can effectively operate at a lower primary fluid exit temperature (up to ∿450 F [505 K]) than steam can.

Gas-Turbine Bottoming Cycle

A gas turbine cycle is not a good bottoming-cycle candidate unless a reasonably high turbine inlet temperature is available to make the cycle efficiency reasonably attractive. However, the gas-turbine cycle is ideally suited to a dry-site application and to a magnetohydrodynamic (MHD) cycle, where the exhaust temperatures are sufficiently high for a gas turbine bottoming cycle. The temperature of fluid leaving the gas turbine is still very high, and additional means need to be employed to extract further energy from the turbine exhaust stream. Only two specific cases of gas turbine bottoming were examined: an open-cycle MHD cycle with air turbine bottoming, and a closed-cycle liquid-metal MHD cycle with helium bottoming cycle.

Both the performance and the cost estimating for the gas-turbine bottoming cycles were based on procedures identical to those explained in the earlier sections and on the open- and closed-cycle gas turbines. Hence these bottoming cycles are not described further here.

ENERGY EXTRACTION LIMITS—PINCH-POINT PROBLEM

In an effort to improve the overall cycle efficiency, the bottoming cycle seeks to extract maximum energy from the prime-cycle exhaust fluid stream (source fluid). Since the energy transfer between source fluid and bottoming-cycle fluid (secondary fluid) occurs in the bottoming-cycle vapor generator, the actual energy extraction is limited by physical and economic considerations of a heat exchanger design. Both the size and the cost of heat exchanger equipment are strongly influenced by a minimum local temperature difference between the source fluid and the secondary fluid occurring anywhere in the heat exchanger. This minimum acceptable temperature difference occurs at a "pinch point" in the heat exchanger.

To see how the pinch point affects the energy extraction from a source fluid, consider a typical situation in which the prime-cycle fluid is a gas, and the bottoming cycle is of Rankine type in which the secondary fluid undergoes a change of phase in the heat exchanger. Steam is a common secondary fluid of this nature. There are two pinch points applicable to such a heat exchanger:

- Superheater Δt, between inlet prime-cycle fluid temperature and exit bottoming-cycle vapor temperature. (A nominal value of 50 F [27.8 K] was specified for this pinch point in the study.)
- Boiler Δt, occurring anywhere outside the superheater section in the vapor generator. (A nominal value of 30 F [16.7 K] was specified for this pinch point.)

Of the above two pinch points, the superheater pinch point is usually not limiting unless the inlet gas temperature is very low.

A typical gas/steam vapor generator is represented in Figure 4-2. The source gas inlet temperature at point A of the figure is set by the prime-cycle design, and the inlet temperature of the secondary fluid at 1 is primarily dictated by the coolant temperature of the bottoming cycle condenser. The temperature-enthalpy relation for the secondary fluid is a function of the type of fluid and the selected cycle conditions (for example, pressure and temperature at the turbine inlet, point 2).

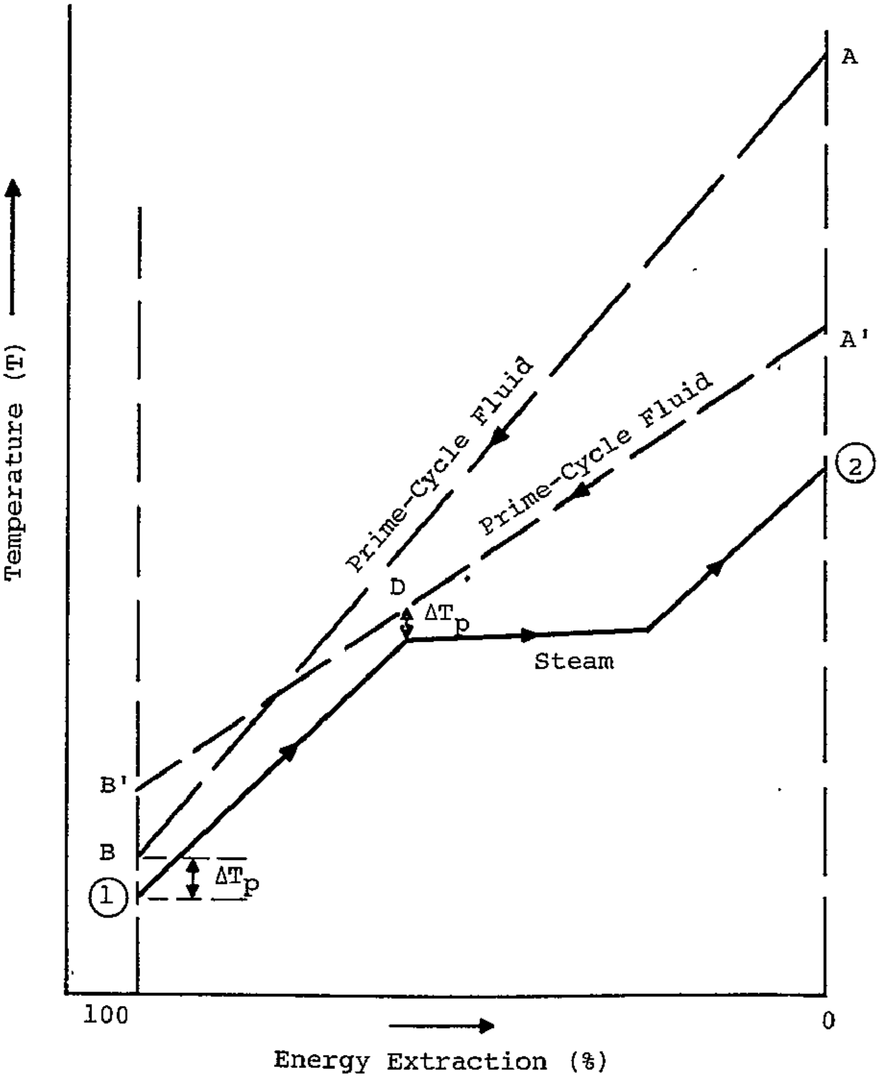

Figure 4-2. Pinch-Point Effects

Assuming constant specific heat for the prime-cycle fluid, the straight-line trajectory of the gas path AB (Figure 4-2) specifies the allowable energy extraction rate. With the temperature at A sufficiently high, the superheat pinch point does not apply and the boiler pinch point occurs at the economizer inlet, point 1; thus the gas temperature at point B maintains a desired pinch point (ΔT_p) with the secondary fluid inlet temperature. This achieves the maximum energy extraction from the prime-cycle fluid.

As the inlet gas temperature decreases, a temperature is reached at which the pinch-point location moves from the economizer inlet toward the saturated liquid point. This results in an increase in the exit gas temperature. For the gas temperature at point A' in Figure 4-2, the pinch point at D causes a substantial rise in the exit gas temperature at B', thereby causing a significant reduction in energy extraction.

The above discussion applies to a non-reheat bottoming cycle. If the source gas passes through the reheat stage of a bottoming cycle before it interacts with the superheater stage, the reduction in gas temperature at the superheater inlet may make the pinch-point problem even worse. It was for this reason that reheat energy for the steam bottoming cycle for the closed-cycle liquid metal MHD case was supplied in the furnace.

For a given temperature level of prime-cycle fluid gas, the incremental reduction in exit gas temperature due to a reduction in the bottoming-cycle fluid pressure is of a secondary nature. Sizable gains can, however, be achieved by selecting a working fluid with more suitable thermodynamic properties. Figure 4-3

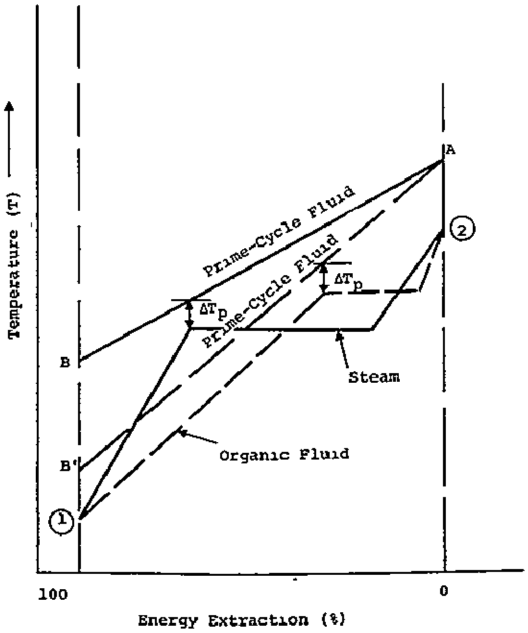

Figure 4-3. Effect of Working Fluid

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indicates the advantages of an organic fluid; for the same end-temperature levels, 1 and 2, and for the same gas inlet temperature and pinch-point temperature difference (ΔT_p), the gas exit temperature at B' for the organic cycle is substantially below the gas temperature at B for the steam cycle (Figure 4-3). This gain is due to lower latent heat of the organic fluid in comparison with steam. It must be noted, however, that the fluid selection criteria also include the temperature limit at which the organic fluid can safely operate in a power cycle without thermal decomposition.

LARGE STEAM BOTTOMING CYCLES

CYCLE SELECTION

Four advanced cycles utilizing the large steam bottoming cycle are described below. In each case, the throttle steam conditions are 3500 psig (24.2×10^6 N/m^2) at 1000 F (811 K), with a single reheat to 1000 F (811 K). The final feedwater temperature (FFWT) is selected from the three options:

- FFWT = 117 F (320 K) (no feedwater heating; deaeration is accomplished in the condenser).
- FFWT = 232 F (384 K) (feed heating train includes one closed heater followed by a deaerator heater).
- FFWT = 510 F (539 K) (complete feed heating train, with five heaters, typical for modern steam power plants).

For a given energy input rate to the steam cycle, the cycle efficiency (heat rate) improves with the increase in recuperation. Thus the increase in final feedwater temperature is accompanied by better steam cycle efficiency, as indicated in Table 4-1.

Table 4-1

STEAM CYCLE EFFICIENCY AS A FUNCTION
OF FEEDWATER TEMPERATURE

(Basis: 3500 Psig/1000 F/1000 F, 1.5-In. Hga
Back Pressure. Approximate Output: 700 MW)

Final Feedwater Temperature (°F)	Cycle Efficiency (%)	Heat Rate (Btu/kWh)
117	37.5	9110
232	40.6	8412
510	45.3	7538

The selection of FFWT for each cycle seeks to maximize energy extraction from a given prime cycle, constrained by the pinch-point limits.

Open-Cycle MHD: Here, FFWT of 232 F (384 K) was selected since this allowed employment of the most efficient steam cycle and still maintained the combustion gas temperature at the minimum stack gas temperature limit. The energy input to the steam cycle occurs in several places (see Section 2.8). The pinch-point problem is thus more easily managed. A selection of the more efficient steam cycle with 510 F (539 K) FFWT would result in excessive stack losses as compared to a cycle with 232 F (384 K).

Closed-Cycle Inert Gas MHD: For the "topping cycle" group, an FFWT of 232 F (384 K) was selected; an FFWT of 510 F (539 K) was chosen for the "parallel cycle" group. Higher FFWT for the parallel cycle group results in a better cycle efficiency, and also leaves enough energy in the exit gas for employment of an air preheater (see Section 2.8).

Closed-Cycle Liquid Metal MHD: An FFWT of 117 F (320 K) was selected here. Because of the lower temperature level of the helium gas entering the steam cycle, the reheat energy was supplied in the primary heat input furnace to work around the severe pinch-point problem.

High-Temperature Fuel Cells: Since the combustion gas leaving the vapor generator is used to preheat the air for the prime cycle, an FFWT of 510 F (539 K) was employed in this case.

PERFORMANCE ESTIMATION AND COMPONENT DESIGN

For each of the above cycle cases, the flow rate and temperatures/enthalpies of the prime-cycle fluid entering a bottoming-cycle heat exchanger were available. The selection of the FFWT, and therefore the temperature of prime-cycle fluid, was consistent with achieving a maximum overall cycle efficiency and maintaining a proper stack gas temperature.

In the three MHD cycles, the drive power requirements for various compressors and pumps are substantial. The bottoming-cycle steam turbines were used as a convenient source of required drive power. For most topping cycles of the closed-cycle inert gas MHD and for all cases of closed-cycle liquid metal MHD, the total steam turbine power output was matched to the drive power requirement. The cycles therefore had no electrical output from the bottoming cycle. For the open-cycle MHD cases, two parallel steam turbine sets were required: one for the compressor drive, and the other for the electric power generation.

For the component sizing and design purposes, three parallel information flows were established:

1. The Large Steam Turbine-Generator Department of the General Electric Company was supplied with the actual data on total energy input (Btu/hr) to the steam cycle and the FFWT for each case. By the use of previously developed analysis procedures, estimates of the required turbine sizes and the corresponding heat rates were obtained in each case. The following ground rules were used in the estimation process:
   - Turbine type: tandem compound with a 33.5-inch last-stage bucket, with two, four, or six low-pressure sections (i.e., TC 2F-33.5, TC 4F-33.5, or TC 6F-33.5) as required.
   - No makeup (% MU = 0).
   - Heat rate to include power requirements for the boiler feedpump drive and generator losses.

   The net output from the steam bottoming cycle is the ratio of total energy input to the steam cycle to the cycle heat rate.
2. The Foster Wheeler Corporation supplied design of the steam generators based on the following data:
   - Prime fluid flow rates and temperatures and pressure at inlet and exit.
   - Pressure, flow, and temperature of the feedwater, superheater exit steam, cold reheat, and reheat return steam.

COMPONENT COST ESTIMATES

The capital cost estimates of the major components were supplied by the respective groups responsible for their design. The procedure used for the steam-turbine-generator cost estimate is identical to that followed for the advanced steam cycle, and the discussion in Section 2 is directly applicable here. The basis for vapor-generator cost estimates is described in Section 6.

MEDIUM STEAM BOTTOMING CYCLES

CYCLE SELECTION

The medium steam bottoming cycles were used with three advanced cycles. In each case, the Medium Steam Turbine-Generator Product Department (MST-GPD) of the General Electric Company was responsible for cycle performance calculations, for the steam turbines, and for the design and cost estimates of the vapor generators (referred to here as heat recovery steam generators [HRSG]) The details of the feedwater heating train and the boiler feedpump were developed by the Bechtel Corporation as balance-of-plant equipment.

The selection of parameters for the steam bottoming cycle is constrained by the allowable pinch-point temperature differences in the heat-recovery heat exchanger and by the lowest allowable temperature of the prime-cycle fluid. A nominal boiler pinch-point temperature difference of 30 F (16.7 K) was assumed, and the effect of temperature differences of 50 F (27.8 K) and 70 F (38.9 K) were examined. A stack gas low-temperature limit of 300 F (422 K) was specified for all combustion product gases.

If a steam supply is required for process purposes in the plant (e.g., for NO_x suppression in gas turbine combustors, for a booster-compressor drive turbine, for the LBtu gasifier), it can be obtained from either of the two sources:

1. Extraction from a steam turbine stage appropriate to the desired steam pressure can be employed. For a reheat cycle, the cold reheat point provides a most convenient point for process steam extraction. The crossover pipe feeding steam to the low-pressure turbine stages is another convenient source of low-pressure steam.
2. If a dual-pressure HRSG design is employed, the process steam can be obtained directly from the HRSG. The low-pressure drum generates saturated process steam, and lowers the prime-cycle fluid exit temperature as well.

The availability of the process steam and a low stack gas temperature are ensured by (1) a non-reheat single-pressure cycle, (2) a non-reheat dual-pressure cycle, and (3) a reheat cycle; for the reheat cycle, a dual-pressure HRSG is not required. For the LBtu cycle cases with a reheat steam bottoming cycle, a fraction of the throttle steam is supplied to the noncondensing steam turbine drive for the booster compressor, and this turbine exhaust is used as a process steam input to the gasifier. For the LBtu cases with a non-reheat steam cycle, the saturated steam generated in the low-pressure drum of the HRSG is supplemented by the steam supply from the turbine system, without employing an extraction point. A fraction of the throttle steam is passed through a high-pressure, noncondensing turbine in parallel with the main high-pressure turbines, and the turbine exhaust serves as a source of supplemental process steam. These cycle arrangements are illustrated in Figures 4-4 through 4-6.

Steam bottoming cycles with various throttle steam conditions, and with and without a reheat stage, have been examined. The actual steam cycle efficiency is significantly influenced by the following features of an integrated combined cycle:

- As described above, the entire high-pressure steam supply does not flow through the condensing turbine. This is due to process steam requirements.
- For LBtu cycles a significant makeup water flow is required, to compensate for the loss of gasifier steam through the stack. The corresponding increase in the

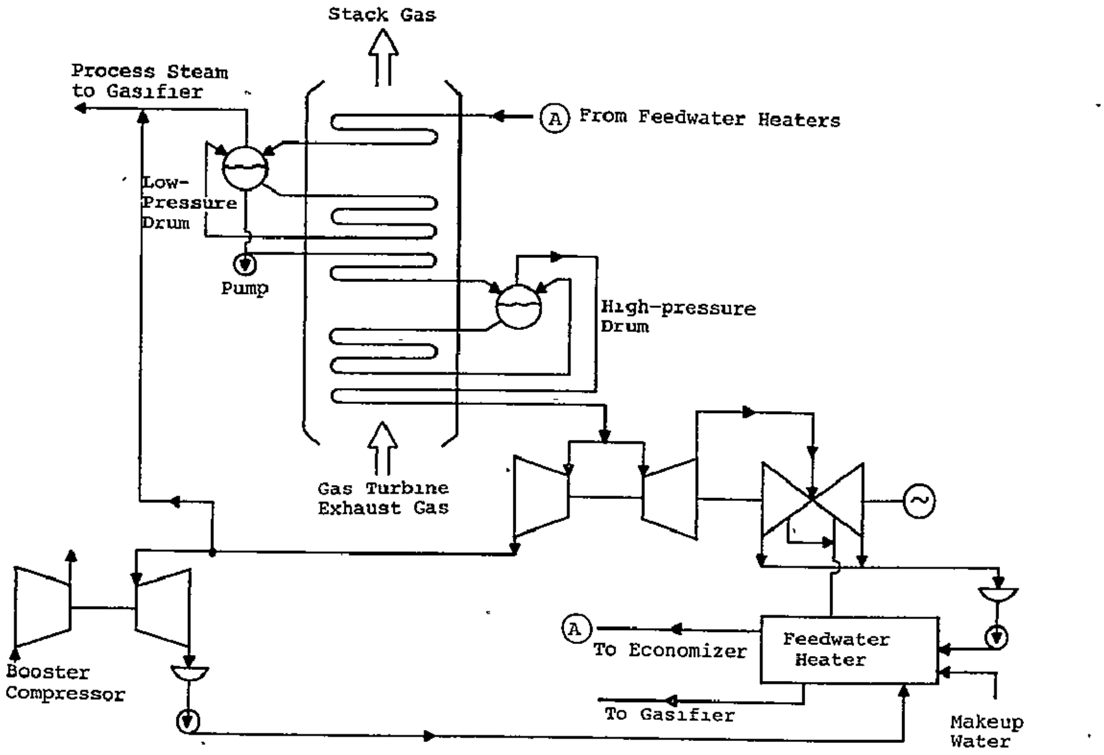

Figure 4-4. Non-reheat Steam Bottoming Cycle for Air-Cooled Combined Cycle—Integrated LBtu Cases

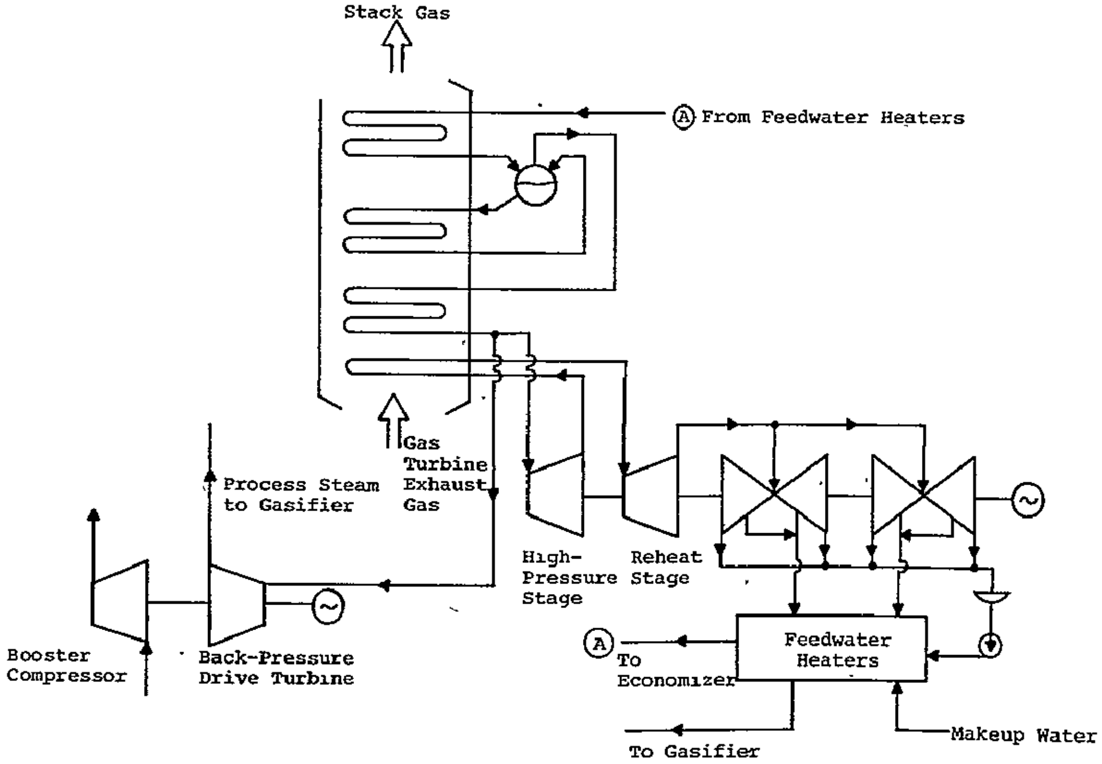

Figure 4-5. Reheat Steam Bottoming Cycle for Air-Cooled Combined Cycle—Integrated LBtu Cases

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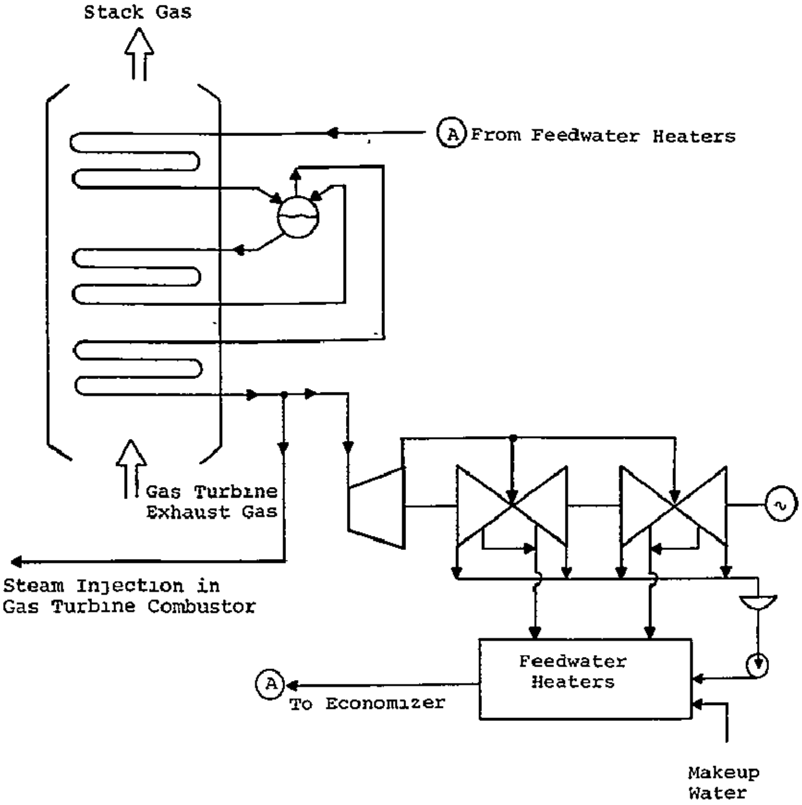

Figure 4-6. Non-reheat Steam Bottoming Cycle for Air-Cooled Combined Cycle—Other Clean Fuels (except LBtu)

feedwater heating duty results in a higher-than-normal extraction steam flow to the feedwater heaters, thereby causing a reduction in the output power.

- The final feedwater temperature is not held constant for all steam bottoming cases. For the air-cooled cases, a nominal final feedwater temperature of 250 F (394 K) is used as a design guide, and the actual temperature is determined by the location of available low-pressure extraction stages, which match the specific turbine selected for each case. For the water-cooled cases, a lower limit of 232 F (384 K) was used for the final feedwater temperature.
- In most water-cooled cases, the energy input from the gas turbine bucket and nozzle cooling loops increased the final feedwater temperature.

These variations make specification of stand-alone steam-turbine cycle efficiency inappropriate and misleading.

Open-Cycle Gas Turbine Combined Cycle—Air-Cooled and Water-Cooled

The base cases and many of the parametric points analyzed in these two cycle categories consumed low-Btu (LBtu) gas supplied from an integrated coal-gasification plant. The gasification plant required a substantial amount of low-pressure saturated steam for the process. A major fraction of this process steam was generated by using a dual-pressure HRSG.

Figure 4-7 is a schematic outline of a dual-pressure heat recovery steam generator (HRSG). Exhaust gas from the gas turbine passes first through the high-pressure steam generation section, then through a low-pressure section. The pressure level in the low-pressure drum is determined by the pressure requirements of the process steam.

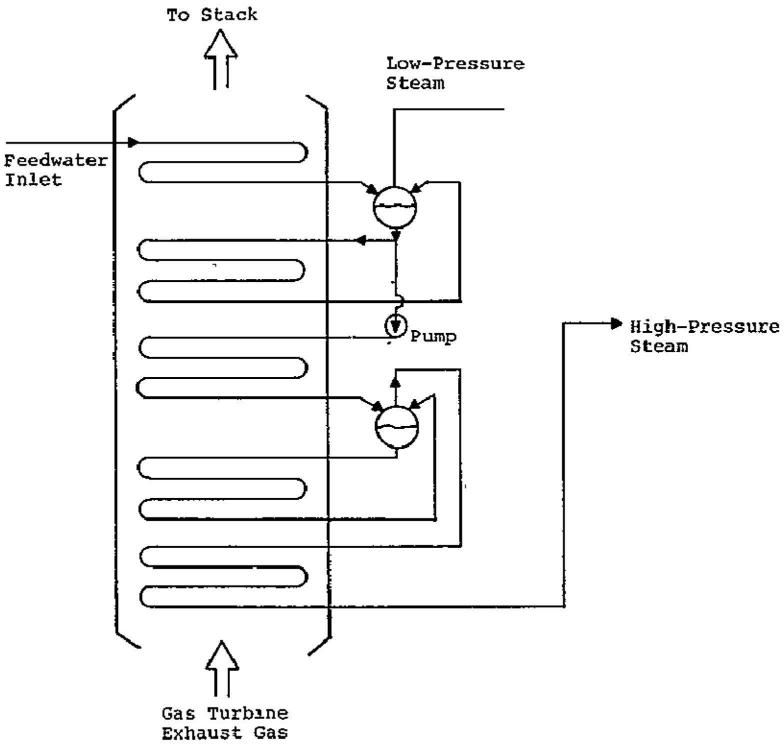

Figure 4-7. Dual-Pressure Heat Recovery Steam Generator

A dual-drum arrangement provides greater energy extraction from a source gas stream than from a single-drum arrangement at the same throttle conditions. As can be seen in Figure 4-8, switching from a single-drum to a dual-drum system causes the stack temperature to decrease from T_x to T_y for the same feedwater temperature. Of course, the pinch-point temperature difference at the low-pressure drum is an additional constraint.

Three different steam cycle configurations were evaluated as steam bottoming plants for the cases of air-cooled combined cycles:

1. Non-reheat Cycle with Integrated LBtu Plant (Figure 4-4). The process steam demand not satisfied by the low-pressure drum generation is supplied from the exhaust from one of the two high-pressure turbines. The steam required by the drive turbine for the gasifier booster compressor is also supplied from this high-pressure turbine. The process

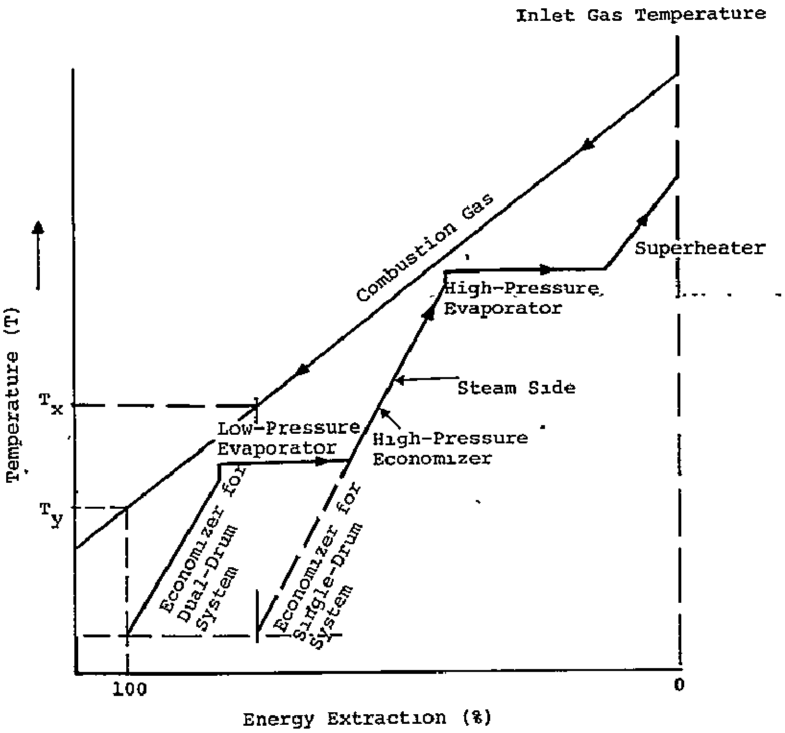

Figure 4-8. Combustion Gas/Steam Temperature Distribution

water requirements of the gasifier were satisfied from the feedwater heating train. Since the process steam and water supplied to the gasifier are lost from the closed steam loop, makeup water is continuously added to the feedwater heating train.

2. Reheat Cycle with Integrated LBtu Plant (Figure 4-5). The reheat HRSG is of a single-drum type. A fraction of the throttle steam is supplied to a back-pressure drive turbine for the booster compressor, and the turbine exhaust serves as external process steam required by the gasifier. In this arrangement, some net electric power is generated at the drive-turbine shaft. The feedwater train is similar to that in configuration 1, above. A few cases of this type were analyzed to provide a basis of comparison with the non-reheat dual-drum cycles.

3. Non-reheat Cycle for Other Clean Fuels (Figure 4-6). This is a single-drum steam cycle of a conventional type. The only interaction of the steam bottoming cycle with the rest of the cycle is to supply steam for injection in the gas turbine combustor for NO_X suppression. Makeup water is added to the feedwater heating train to account for the loss of injected steam.

In addition to these three cycle configurations, one case each from configurations 1 and 2 was evaluated with the supplemental firing in the HRSG. In these cases, a small amount of LBtu gas is burned in the gas turbine exhaust to raise its temperature to 1400 F (1033 K) before it enters the HRSG. (Since the supplemental firing raises the temperature of the prime-cycle fluid entering the HRSG, the pinch-point problem is greatly simplified by pushing the pinch point toward the economizer inlet.)

The nominal gas turbine exhaust temperature for the air-cooled cases was about 1100 F (866 K). Hence, the throttle steam conditions of 1250 psig (8.7×10^6 N/m^2) and 950 F (783 K) were specified for the steam cycle. A feedwater heating train consisting of one closed heater and one deaerator-heater was used for all cases, with a nominal FFWT of 250 F (394 K). Parametric cases were examined with a throttle pressure range of 1000 to 1800 psig (7×10^6 to 12.5×10^6 N/m^2) and throttle temperature range of 900 to 1000 F (755 to 811 K), to provide a basis for comparison. When reheat turbines were considered, a reheat at 950 F (783 K) was utilized.

The three steam cycle configurations used with the air-cooled gas turbine cycle were also considered with the water-cooled cycle. No supplemental fired cases were considered. The feedwater train is more complex in this cycle. The gas turbine nozzle coolant forms a closed fluid path with the feedwater heating train. The bucket coolant also transfers significant thermal energy to the feedwater train by means of a heat exchanger. The net thermal energy added from the gas turbine coolant streams to the feedwater train is, in most cases, enough so that no additional steam extraction from the turbines is required.

Closed-Cycle Gas Turbine

The helium gas temperature entering the bottoming-cycle heat exchanger was in the range of 463 to 967 F (513 to 793 K). Because of the low inlet temperature of the prime-cycle fluid, the throttle steam temperature was governed by the minimum gas-to-superheat temperature difference limit. A steam pressure appropriate to the throttle temperature selected in each case is given in Table 4-2. No feedwater heaters were used.

Table 4-2

STEAM BOTTOMING CYCLE PERFORMANCE
FOR CLOSED-CYCLE GAS TURBINE CASES

Case	Prime-Cycle Fluid Inlet Temperature (°F)	Throttle Steam Condition (psig/°F)	Steam Cycle Efficiency (%)
42	463	100/413	19.9
43	463	100/413	16.2
44	611	400/561	26.4
45	434	125/384	20.2
46	967	800/900	32.7

Note: Turbine back pressure for Case 43 = 5 in. Hga; all others 1.5 in. Hga.

The steam cycle is shown in Figure 4-9. The heat recovery steam generator (HRSG) is of shell-and-tube design. Because of the low steam pressures, only low-pressure turbine units were required.

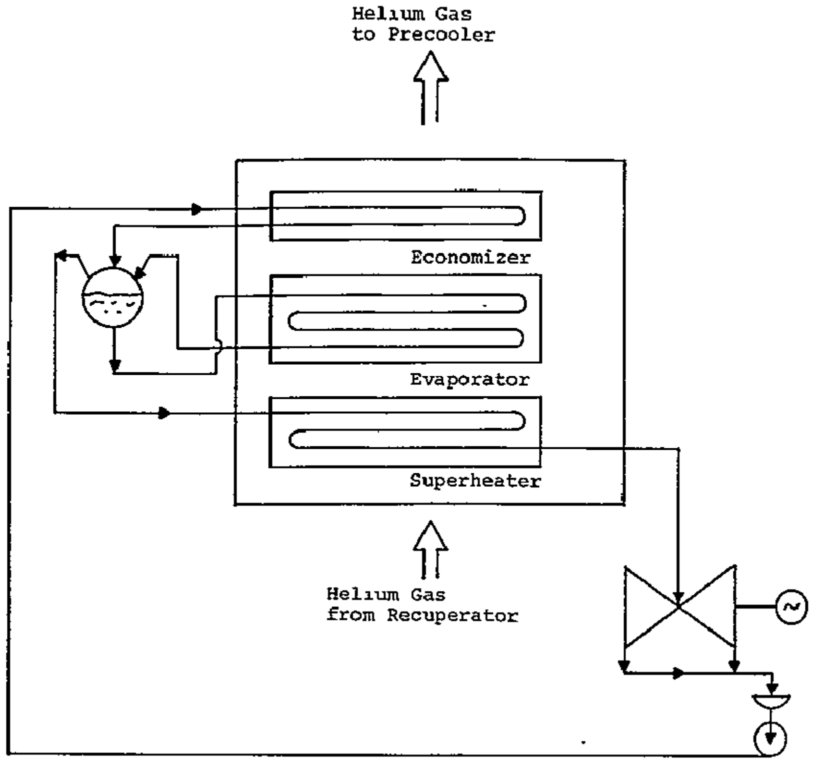

Figure 4-9. Non-reheat Steam Bottoming Cycle for Closed-Cycle Gas Turbine

PERFORMANCE ESTIMATION AND COMPONENT DESIGN

For the air-cooled and water-cooled combined cycle cases, the following data were available from the prime-cycle advocate's analysis:

- Exhaust gas flow rate, temperature, and enthalpy
- Temperature-enthalpy data for the exhaust gas down to 300 F (422 K)
- Allowable gas-side pressure drop (in. H_2O) in the HRSG
- For LBtu fuel cases:

  Gasifier process steam flow/pressure requirements
  Booster-compressor drive turbine power and corresponding steam flow requirements
  Gasifier process water requirements from the deaerator feedwater heater

- For other fuel cases:

  Steam injection flow/pressure for gas turbine combustor

For the water-cooled cases, the following additional data were supplied:

> Transition piece and nozzle coolant flow, temperature, enthalpy, and pressure from and to the deaerator heater, including the steam production in a flash tank
>
> Bucket coolant flow, temperature, enthalpy, and pressure from the condenser tank, recovered flow, and makeup flow

The above data, along with the cycle conditions described in Figures 4-4 through 4-6, formed the basis for performance estimation by the Medium Steam Turbine-Generator Product Department. In each case, the HRSG was first designed to estimate throttle steam flow and low-pressure drum flow, if any, for given prime-cycle fluid conditions. The steam turbine cycle was designed with provision for the process steam and water flow demands, and net electrical power generation was estimated. The analysis procedures used in these designs were those employed for commercial unit designs, and reflect practical design and manufacturing considerations.

In the case of the closed-cycle gas turbine, the flow rate/temperature of the helium gas leaving the regenerator and helium temperature at the compressor inlet were specified. The steam bottoming cycle was selected to achieve helium exit temperature as close to compressor inlet condition as possible. The shell/tube design of HRSG was developed jointly by the Medium Steam Turbine-Generator Products Department (MSTGPD) and the Heat Transfer Products Department (HTPD) of the General Electric Company.

COMPONENT COST ESTIMATES

The capital cost estimates of open-cycle HRSG units were developed by MSTGPD, and those for closed-cycle cases were jointly developed by MSTGPD and HTPD. The steam turbine costs were based on standard General Electric pricing policy, similar to the pricing policy used by the Large Steam Turbine-Generator Department, as discussed in detail in Section 2.6.

ORGANIC BOTTOMING CYCLES

CYCLE SELECTION

The organic bottoming cycles were evaluated with two types of advanced cycles:*

> Open-Cycle Recuperative Gas Turbine Cases: In the cases examined in this study, the inlet temperature of prime-cycle gas (combustion gas) was in the range of 700 to 828 F (644 to

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*The Thermo Electron Corporation, under subcontract with the General Electric Company, was responsible for the cycle configuration, for the selection of an organic working fluid, and for the cycle performance calculations.

715 K). The organic working fluid selected for this application was Fluorinol-85 (Fl-85) with the boiler outlet conditions of 600 F (589 K) and 700 psia (4.8 x 10^6 N/m^2), operating in a subcritical cycle. Figure 4-10 is a schematic flow diagram for the organic bottoming cycle. The allowable minimum temperature for the combustion gas, and hence for the liquid entering the vapor generator, was limited by the dew-point considerations in the exhaust gas. Since the turbine expansion for Fluorinol-85 occurs solely in the vapor superheat region, a regenerator was used. The use of a recuperator results in higher exhaust gas temperature and avoids the problem of exhaust stream dew point.

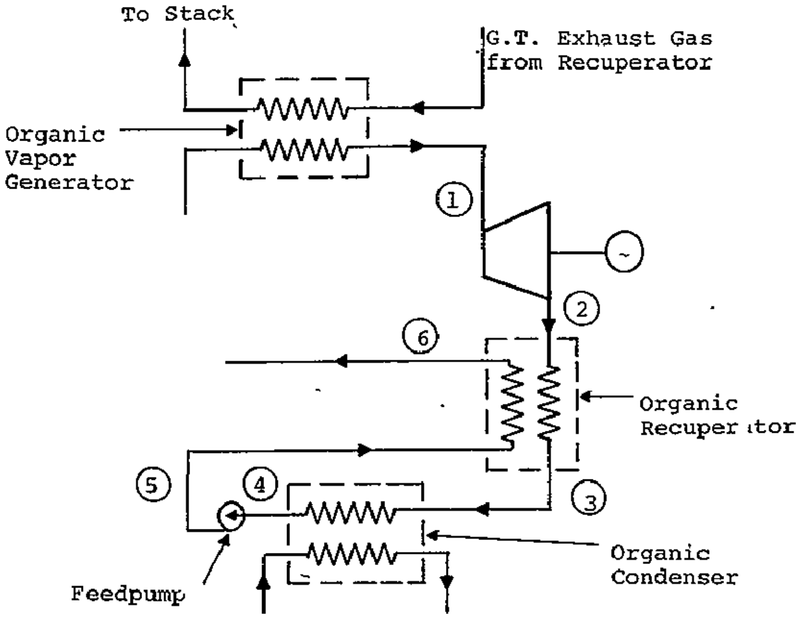

Figure 4-10. Organic Bottoming Cycle for Open-Cycle Gases

<u>Closed-Cycle Gas Turbine Cases:</u> Except for a single case with a prime-cycle fluid inlet temperature of 611 F (595 K), a helium inlet temperature in the vicinity of 450 F (505 K) was typical for the bottoming cycles evaluated with this prime cycle. In these cases, the helium gas is cooled to the lowest possible temperature, to maximize the heat recovery and to minimize the pre-cooler requirements. For prime-cycle fluid temperature of 611 F (595 K), Fluorinol-85 operating in a subcritical cycle was used; for the other cases, Refrigerant 22 (R-22) operating in a supercritical cycle was selected. Figure 4-11 is a schematic cycle diagram used for R-22 and for Fluorinol-85 without a recuperator.

For all of these cases, calculations were made with a variety of boiler outlet pressures and temperatures before the selection was made. Calculations for the case of Fluorinol-85 and 611 F (595 K) gas temperature indicated that the optimum cycle occurred with saturated vapor outlet conditions, and that raising the boiler outlet temperature for this case reduced the power output.

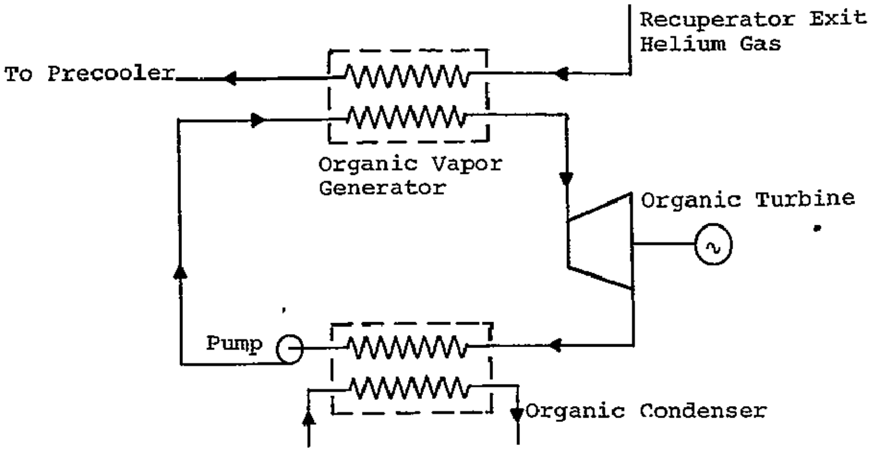

Figure 4-11. Organic Bottoming Cycle for Closed-Cycle Gases

One area of practical concern with the use of R-22 is the higher pressure of R-22 relative to the pressure of the helium gas. Leakage of R-22 into the helium circuit must be prevented because the peak temperature in the gas turbine is far above the thermal decomposition limit of R-22.

FLUID SELECTION PROCEDURE

Thermodynamic Considerations

In an effort to evaluate a candidate working fluid on the basis of thermodynamic considerations, the maximum work obtainable with a thermodynamically perfect bottoming plant was employed as a standard of comparison for the actual plant.

Consider the reversible, open-circuit, work producing device illustrated in Figure 4-12, in which exhaust gases are taken into the device at initial state 1 and delivered from the device at the final state 2. The change from state 1 to state 2 (Figure 4-12) of the gas is accomplished by means of one or more reversible cyclic heat engines operating between the local gas temperature and ambient temperature in such a way that complete reversibility is maintained. This requires that all heat transfer between (1) the exhaust gas and cyclic heat engine and (2) the cyclic heat engine and environment occur with zero temperature difference, as indicated in Figure 4-12. The work obtained in this fashion represents a unique maximum quantity produced by an open-circuit device which can exchange heat only with an isothermal energy reservoir, the environment. This maximum obtainable work is termed the "available energy" and requires only that all processes that bring about the given change of state be reversible.

Open-Cycle Recuperative Gas Turbine Cases. Figure 4-13 illustrates the thermodynamic cycle used for the Fluorinol-85 working fluid. The state points indicated correspond to similar points shown in Figure 4-11. Note that recuperation reduces the vapor

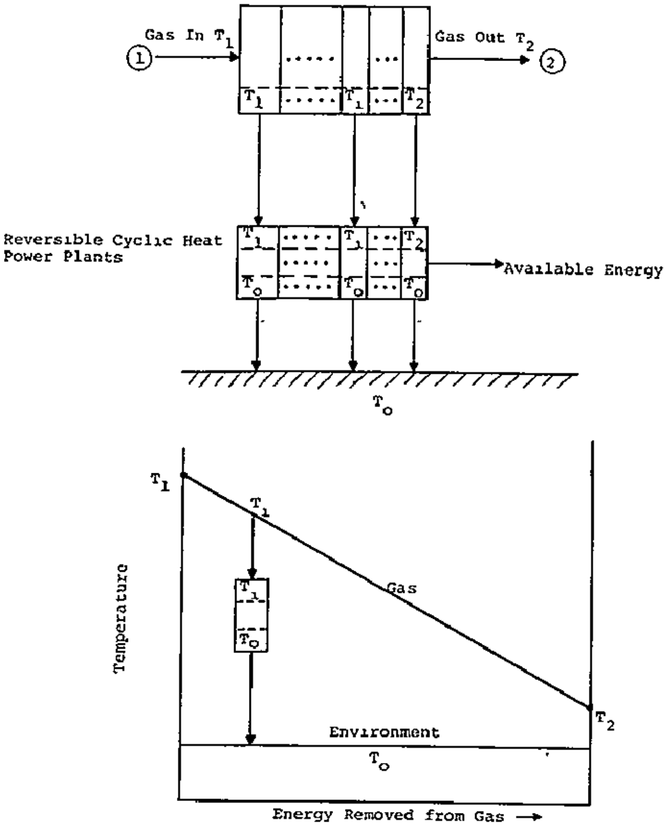

Figure 4-12. Reversible, Open-Circuit, Work Producing Device

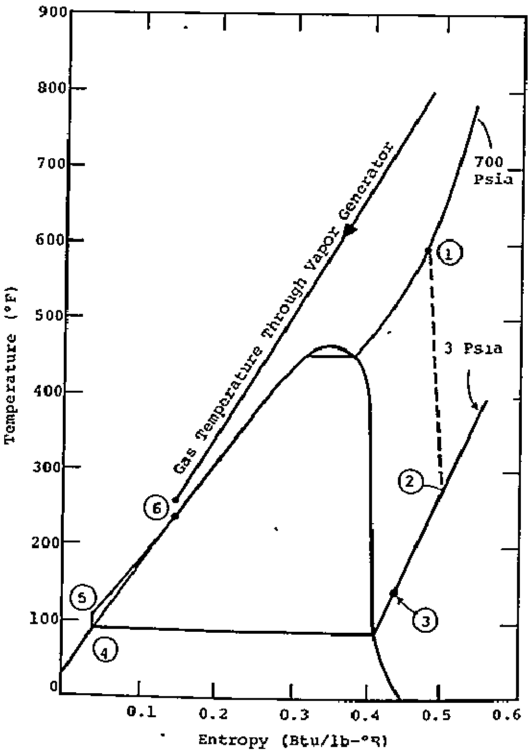

Figure 4-13. Temperature-Entropy Diagram for Fluorinol-85 Bottoming Cycle

temperature to near saturation before it enters the condenser and preheats the vapor-generator feed liquid above 200 F (366 K), thus preventing dew-point corrosion in the vapor generator. The recuperation still permits extraction of the maximum heat from the gas heat source. Further, the countercurrent heat transfer between the prime-cycle fluid and the Fl-85 working fluid, state points 6 to 1 in Figure 4-13, occurs with a relatively small variation in the gas-to-working-fluid temperature difference at any point in the vapor generator. The pinch point occurs at the liquid inlet to the vapor generator.

For Fl-85, the best performance is obtained with heat source temperatures in the range of 600 to 750 F (589 to 672 K). For each of the open-cycle cases, the ratio of the actual work output to the reversible work output is given in Table 4-3, and it varies from 0.63 to 0.65. In these calculations, the ambient temperature is taken as the condensing temperature in order to eliminate the condenser temperature difference from the comparison. The factors which account for the reduction of this ratio below one are:

- Turbine losses
- Feedpump losses } ∿16 percent work loss
- Loss due to vapor-generator temperature difference
- Line pressure losses } ∿20 percent work loss

These losses might be reduced somewhat by

a. Using a different working fluid, with higher thermal stability and operating on a supercritical cycle to reduce the vapor-generator temperature difference. No such working fluid is available at present.

b. Increasing the boiler outlet temperature with Fluorinol-85 to 650 F (616 K), which capsule testing indicates may be satisfactory.

In any case, the maximum improvement that might be achieved is small, because in a practical design the vapor-generator temperature difference loss cannot be reduced below approximately 10 percent, even with a perfect fluid.

Closed-Cycle Gas Turbine Cases. Figure 4-14 is a temperature enthropy diagram for the bottoming cycle using R-22. A recuperator is not required here. It is seen that the vapor-generator temperature difference is remarkably constant, and that the turbine exhaust vapor has a relatively small superheat. As indicated in Table 4-3, the ratio of the actual work output to the reversible work output is in the range of 0.52 to 0.60 for all of the R-22 cases. This relatively low ratio is the result of two factors:

- The effect of a given vapor-generator temperature difference becomes more important as the temperature of the prime-cycle fluid decreases.

Table 4-3

SUMMARY OF SHAFT POWER OUTPUT AND OTHER PARAMETERS FOR ORGANIC BOTTOMING CYCLES

Case Number	Working Fluid	Boiler Outlet		Condenser Pressure (psia)	Organic Flow Rate (gal/min)	Gas Temp (°F)		Reversible Power (hp)	Actual Power / Reversible Power
		P (psia)	T (°F)			To Vapor Generator	From Vapor Generator		
Open Cycle									
30,32,33	Fl-85	700	600	2.9	1,630	827.6	256.4	51,400	0.646
31	Fl-85	700	600	2.9	1,580	808	256.4	49,064	0.653
34	Fl-85	700	600	2.9	1,580	827.6	276.4	50,419	0.635
35	Fl-85	700	600	2.9	1,520	827.6	296.4	49,359	0.626
36	Fl-85	700	600	2.4	1,630	827.6	249.9	52,580	0.651
Closed Cycle									
34,35,36	R-22	1700	430	188.4	23,210	463	135.5	154,771	0.596
37	R-22	1600	410	188.4	22,600	463	154.3	151,801	0.555
38	R-22	1500	390	188.4	21,940	463	173.1	148,900	0.517
39	R-22	1700	430	208.0	23,530	463	142.4	148,315	0.585
40	Fl-85	650	460	2.4	11,680	611	142.4	272,961	0.707
41	R-22	1500	400	188.4	22,140	434	133.1	134,507	0.599

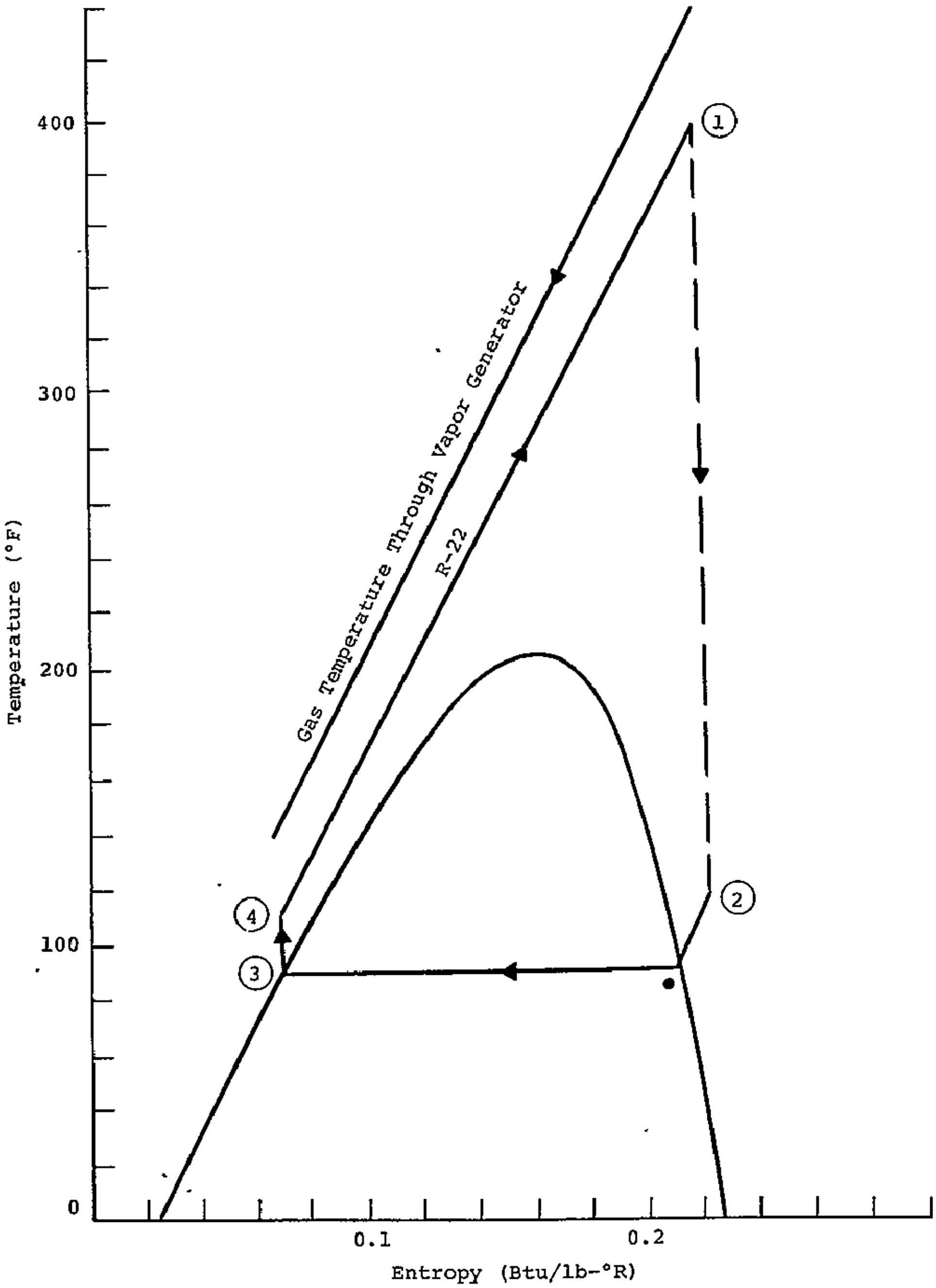

Figure 4-14. Temperature-Entropy Diagram for Bottoming Cycle, Using Refrigerant-22

- The feedpump work is a relatively large fraction of the gross turbine output; hence the feedpump inefficiency has a relatively large effect on the overall system performance.

However, with the low-temperature prime-cycle fluid it is doubtful that any substantial improvement will be achieved by the use of any alternative working fluid, though some improvement may be possible. For the Fl-85 cycle with higher prime-cycle temperature, an actual-to-reversible power ratio of 0.71 was achieved.

Other Selection Criteria

With the thermodynamic considerations discussed above, the following additional criteria were used in the selection of organic working fluids for the power plant application:

- Thermal Stability—demonstrated at boiler outlet temperature in contact with low-cost materials of construction such as carbon steel
- Safety—fire-resistant or nonflammable and relatively nontoxic.
- Corrosiveness—noncorrosive with low cost materials of construction such as carbon steel.
- Cost and Availability—commercially available at reasonable cost.

These and other major physical characteristics of both organic fluids are listed in Table 4-4.

Because the thermal stability considerations are of critical importance, extensive tests have been conducted on both working fluids in capsule-test form and in a simulated power system. For

Table 4-4

ORGANIC FLUID CHARACTERISTICS

Fluid	Fluorinol-85 [85 mole % CF_3CH_2OH 15 mole % H_2O]	Freon-22 [$CHClF_2$]
Freezing point	-82 F	-256 F
Molecular weight	87.7	87
Critical temperature	452 F	205 F
Critical pressure	800.1 psia	721.9 psia
Fluid cost	\$7.50/lb (development quantity \$0.90-\$1.35/lb (estimated cost in volume production)	∿\$0.65/lb
Toxicity	Not classified as toxic via dermal or inhalation pathways	Very low
Flammability	Fire-resistant	Nonflammable
Thermal stability	Demonstrated at 600 F boiler outlet temperature in simulated power system	Acceptable stability to 430 F

example, thermal stability of R-22 was evaluated in the following manner:

a. Capsule tests were conducted to determine the stability of R-22 and Suniso lubricant as a function of time, temperature (3 levels: 425 F [492 K], 450 F [505 K], 500 F [533 K]), metal type (carbon steel, cast iron, copper), and metal surface area (exposed area in the range of 0.5 in.^{2} to 15 in.^{2} [3 x 10^{-4} to 9.6 x 10^{-3} m^2]). Test results indicated a linear dependence of decomposition on exposed surface area; the relative tendency to promote decomposition was least for copper, followed by carbon steel, and then cast iron. For the 425 F (492 K) to 450 F (505 K) range, the decomposition rate corresponds to the usually predicted factor of double to triple for an 18 F (10 K) increase in temperature.

b. The boiler duty-cycle test loop included a control system that permitted operation of the loop on a 45-minutes-on/15-minutes-off basis. The loop was operated for over 600 hours at a pressure of 375 psia (2.6 x 10^6 N/m^2) and a temperature of 425 F (492 K), and for over 1000 hours at 950 psia (6.5 x 10^6 N/m^2) and 425 F (492 K). During these tests, the average estimate of decomposition rate was much less than one percent for 10,000 hours of operation, in spite of the fact that the boiler outlet temperature at times exceeded 435 F (497 K).

On the basis of the above test data, a maximum boiler outlet temperature of 430 F (494 K) was made for R-22. Additional details of the stability tests are available from a Thermo Electron report (ref. 1).

Based upon similar test data, and also confirmed by tests on prototype systems, the thermal stability of Fl-85 at the boiler outlet condition of 600 F (589 K)/700 psia (4.8 x 10^6 N/m^2) is considered satisfactory.

PERFORMANCE CALCULATIONS AND COMPONENT DESIGN

The cycle performance calculations were based on the component efficiencies and system pressure drops given in Table 4-5.

The original performance estimates were based on the turbine thermal and mechanical efficiency values of 0.85 and 0.95, respectively, for both open- and closed-cycle cases. When the preliminary turbine designs were reviewed it was concluded that the efficiency values given in Table 4-5 should be used in estimating electric power output from the bottoming cycle.

TURBINE DESIGN

Preliminary turbine characteristics were developed by the Thermo Electron Corporation for three base cases: open-cycle case

Table 4-5

ORGANIC CYCLE ASSUMPTIONS

	Open Cycle	Closed Cycle
Organic turbine		
Thermal efficiency	0.80	0.80 (FL-85 case) 0.83 (R-22 cases)
Mechanical efficiency	0.98	0.98
Generator efficiency	0.98	0.98
Feedpump efficiency	0.70	0.70
$\frac{\Delta p}{p}$ for vapor generator	0.10	0.05
$\frac{\Delta p}{p}$ for condenser/recuperator	0.10	0.05
Regenerator efficiency	0.85	None

30 and closed-cycle cases 34 and 40. The methods used in the designs were based on previous detailed analyses. The detailed blade profile analysis is defined in two reports (refs. 2 and 3).

The turbine speed is restricted to a minimum of 1800 rpm to permit use of an 1800-rpm generator without gears. The turbine characteristics for the three base cases are summarized in Tables 4-6 through 4-8. In general, these turbines are characterized by several impulse stages followed by a last reaction stage. The stepup in pitch diameter between adjacent stages is significant, resulting in an unusual turbine design.

The turbine materials are not a critical consideration for organic turbines. The maximum turbine temperature is low (≤600 F [589 K]), the operating stress level is low (maximum blade tip velocity = 483 to 945 ft/s (147 to 288 m/s), the working fluids are noncorrosive, and the turbine expansion occurs solely in the superheated vapor range.

COMPONENT COST ESTIMATES

Following are the bases for capital cost estimates of major components:

- Organic Turbine-Generators: The cost of an organic turbine was estimated on the basis of preliminary turbine designs already described. Since a detailed manufacturing cost estimate was outside the scope of the Task 1 activity, the cost estimate was based on extrapolation from two smaller turbines (120 kW and 1100 kW power outputs) for which a detailed manufacturing cost

Table 4-6

ORGANIC TURBINE DESIGN CHARACTERISTICS
(Open-Cycle Case No. 30)

Working fluid	Fluorinol-85
Fluid conditions	700 psia-600 F; 3.42 psia
Shaft power	34,270 hp
Fluorinol-85 flow	1,088,000 lb/hr
Number of turbines	1
Speed	1800 RPM
Number of stages/turbine	5

Stage Characteristics	Stage 1	Stage 2	Stage 3	Stage 4	
Pressure ratio	2	2.2	2.9	3.93	
Type	Impulse 23% partial admission	Impulse 40% partial admission	Impulse full admission	Impulse full admission	
$(\Delta h)_s$, Btu/lb	13.6	16.24	21.6	25.8	
η_{TS}	0.68	0.71	0.755	0.815	
η_{TT}	0.72	0.74	0.805	0.85	
Pitch diameter, inches	46	54	61.5	71	
Blade height, inches	2.4	2.4	2.4	5	
Relative Mach number	0.69	0.715	0.828	0.891	
Velocity ratio	0.42	0.45	0.45	0.46	

Overall Turbine Characteristics

Efficiency (without mechanical losses)	0.80
Weight	63,500 lb
Height (to exhaust flange)	9 ft
Length (no shaft extension)	15 ft
Width	7.5 ft
Inlet line diameter	9 in.
Exhaust line diameter	6.8 ft

Table 4-7

ORGANIC TURBINE DESIGN CHARACTERISTICS
(Closed-Cycle Case No. 34)

Working fluid	R-22
Fluid conditions	1700 psia-430 F; 197 psia
Shaft power	122,000 hp
R-22 flow	13,110,000 lb/hr
Number of turbines	1
Speed	1800 RPM
Number of stages/turbine	3

Stage Characteristics	Stage 1	Stage 2	Stage 3
Pressure ratio	2	2.06	2.06
Type	Impulse admission	Impulse admission	~40% reaction
$(\Delta h)_s$, Btu/lb	10	10.1	9.9
η_{TS}	0.78	0.81	0.83
η_{TT}	0.825	0.845	0.85
Pitch diameter, inches	39	48	56
Blade height, inches	2.4	3.6	5.5
Relative Mach number	0.67	0.60	0.514
Velocity ratio	0.41	0.49	0.57

Overall Turbine Characteristics

Efficiency (without mechanical losses)	0.825
Weight	19,500 lb
Height (to exhaust flange)	6 ft
Length (no shaft extension)	7 3/4 ft
Width	5 ft
Inlet line diameter	18 in
Exhaust line diameter	2.5 ft

Table 4-8

ORGANIC TURBINE DESIGN CHARACTERISTICS
(Closed-Cycle Case No. 40)

Working fluid	Fluorinol-85
Fluid conditions	650 psia-460 F; 2.6 psia
Shaft power	199,900 hp
Fluorinol-85 flow	7,816,000 lb/hr
Number of turbines	4
Speed	1800 RPM
Number of stages/turbine	2

Stage Characteristics	Stage 1	Stage 2
Pressure ratio	15.81	15.81
Type	Impulse 25% partial admission	~40% reaction
$(\Delta h)_s$, Btu/lb	41.43	41.0
η_{TS}	0.73	0.83
η_{TT}	0.78	0.86
Pitch diameter, inches	75.4	107
Blade height, inches	9.8	12.9
Relative Mach number	1.61	1.24
Velocity ratio	0.358	0.52

Overall Turbine Characteristics

Efficiency (without mechanical losses)	0.80
Weight	67,000 lb
Height (to exhaust flange)	10.5 ft
Length (no shaft extension)	14 ft
Width	9.5 ft
Inlet line diameter	10.5 in
Exhaust line diameter	9 ft

estimate was developed. The extrapolated cost numbers were refined to allow for the differences in design conditions. The cost of an 1800 RPM generator was added to these turbine costs.

- Organic Feed Pumps: The selection and cost of feedpumps were based on vendor quotes received for previous work and on catalog information for standard pumps suitable for organic fluids. Multiple-stage centrifugal pumps with high efficiency are available commercially at reasonable cost.
- Working Fluid Inventory and Cost: The estimated working fluid inventory and initial-fill cost are summarized in Table 4-9, based on $1/lb for Fluorinol-85 and $0.60/lb for Refrigerant-22.

Table 4-9

WORKING FLUID INVENTORY AND COST

	Inventory		Cost		
Case No.	Gallons	Pounds	$	$/kwe	Working Fluid
30 Open-cycle	15,500	172,000	172,000	7.1	Fluorinol-85
34 Closed-cycle	40,000	385,000	231,000	3.7	Refrigerant-22
40 Closed-cycle	90,000	1,000,000	1,000,000	7.1	Fluorinol-85

CRITIQUE ON ORGANIC TURBINE DESIGN

The preliminary organic turbine designs, as given in Tables 4-6 through 4-8, were reviewed by a component technology specialist of the General Electric Company. His comments, which follow, were developed on this preliminary design.

Table 4-10 summarizes the findings of the reviewer. Figure 4-15 helps visualize the geometry proposed for each turbine case. These turbines are unusual because of:

a. Very low centrifugal stress levels. It is indicated that they will tolerate operation without overspeed protection, which would mean tolerance to double speed and four times stress. Since Table 4-10 approximates the bucket root centrifugal stress, agreement is reached with the design intent.

b. Increasing root diameters. Large pressure ratio per stage gives large stepups in height from one stage to the next. The increasing diameter was probably selected to ease the stepup problems, but it appears a more conventional solution should be considered, such as constant pitch diameter. No claim was made, however, that this is an optimized design.

Table 4-10

SUMMARY OF ORGANIC FLUID TURBINE DESIGNS

Characteristic	Open Cycle Case 30	Closed Cycle	
		Case 34	Case 40
Fluid	Fl-85	R-22	Fl-85
T_F	600	430	460
Base cycle power, kW	94,000	300,000	300,000/4
Heat source/T_{max}	Gas/828	He/463	He/611
Organic cycle power/ base cycle power	0.26	0.23	0.48
Power			
Flow rate, pps	302.20	3642	2171/4=542.75
Net horsepower	33,200	92,270	193,000/4=48,250
BFP horsepower at 70% pump efficiency	1,068	29,725	6,916/4
Mechanical loss	1,804	6,420	10,527/4
Aero horsepower	36,072	128,420	210,438/4
Cycle assumptions			
Net work, Btu/lb	77.67	17.92	62.85
BFP work	2.50	5.77	2.25
Mechanical loss	4.22	1.25	3.43
Aero work = Eu	84.39	24.94	68.53
$\Sigma\ \Delta h_{isen}$ (aero work and 85%η)	99.28	29.34	80.62
Design evaluation based on stage design			
$\Sigma\ \Delta h_s\ \eta_{ts}$	79.19	24.20	64.27
$\Sigma\ \Delta h_s\ \eta_{tt}$	82.67	25.20	67.57
Σ leaving loss reduction (Eu-$\Sigma\ \Delta h \eta_s$)	5.20	0.74	4.26
Recovery factor required (all stages)	1.49	0.74	1.29
η_t for zero recovery of leaving loss	0.80	0.82	0.80
Adjusted cycle assumptions			
η_t for conservative cycle calculation	0.80	0.82	0.80
η_t for 100% leaving loss recovery	0.83	0.85	0.84
Organic cycle power/ base cycle power	0.24	0.22	0.45
Bucket root centrifugal stress, psi [$A_{AN}(\mathrm{RPM}/1000)^2$ for last stage]	9509	3135	14049
Product cost, $/kW	25.13	10.00	20.13

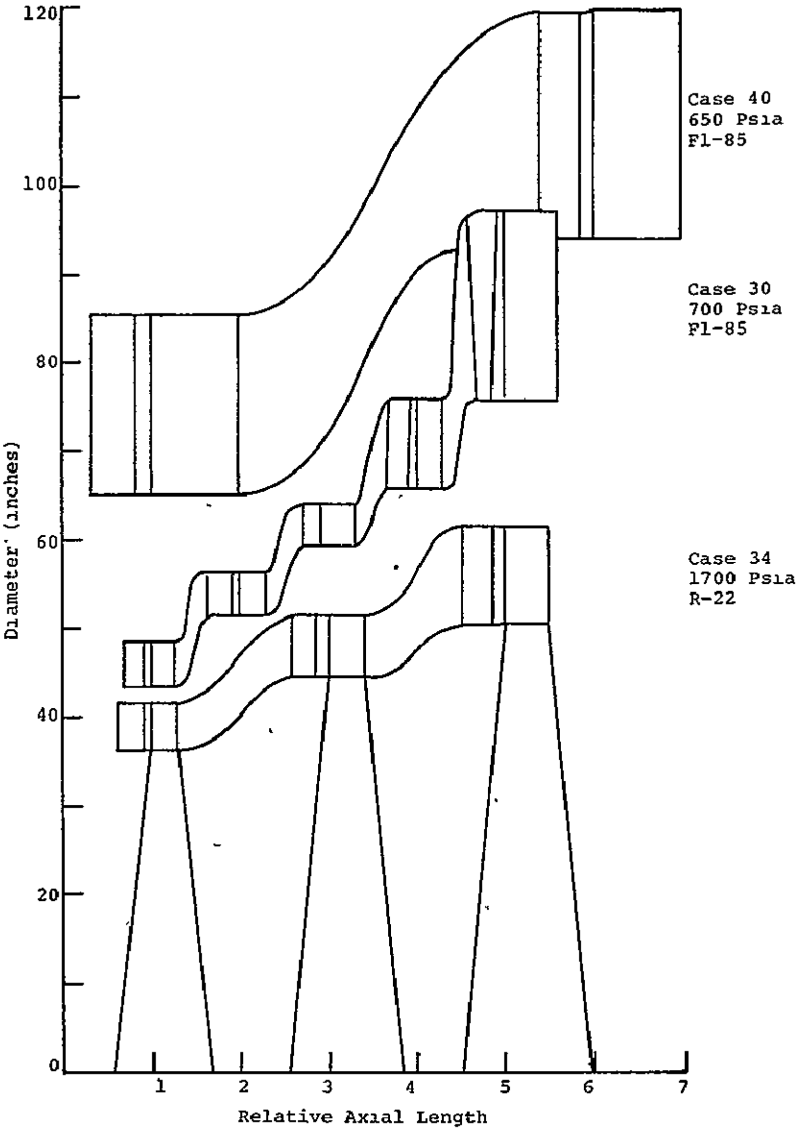

Figure 4-15. Organic Fluid Turbines (1800 RPM)

The preliminary performance predictions have been compared with the cycle analysis assumptions. If cycle assumptions are to be kept conservative, they should be adjusted to the "no recovery of leaving loss" basis, which is considerably lower than the assumed 85 percent overall turbine efficiency. This assumes the efficiency predictions were based on Baljé's papers, presented to the American Society of Mechanical Engineers (refs. 4 and 5), and were appropriately applied. Alternatively, the turbine geometry might be changed for more optimum turbine efficiency. This would bring the predicted turbine efficiency up closer to the assumed 85 percent value for two moderate-pressure-level Fluorinol-85 turbines.

The R-22 case with 1700 psia (11.7×10^6 N/m^2) inlet and 197.8 psia (1.4×10^6 N/m^2) exhaust will suffer some from structural compromise. This turbine is not extreme in radical step-up as suggested by Figure 4-15. For these reasons, an overall efficiency level of about 83 percent is recommended if further studies of

R-22 fluid turbines are to be made. In view of the partial arc feature, an efficiency of 80 percent for case 30 is recommended. For case 40, the efficiency value should be reduced to about 75 percent, to allow for the high Mach numbers and partial arc.*

The last item on Table 4-10 is the cost estimate for each of the turbine elements only. This cost is computed from the cycle calculation of net shaft power, after deducting some substantial accessory and bearing losses, and then deducting another 2 percen for electrical losses of the generator. These costs seem reasonable for a first order approximation, bearing in mind the low efficiency levels of these three machines.

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4. Baljé, O.E., "Axial Cascade Technology and Application to Flow Path Designs," Parts I and II, Transactions of the ASME, Journal of Engineering for Power, Vol. 90, 1968, pp. 309-328 and 329-340.

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*An 80 percent thermal efficiency was used for case 40 in the final cycle calculations, instead of the 75 percent value recommended by the reviewer, because the Thermo Electron Corporation felt they could redesign the turbine with higher (80%) efficiency without appreciable increase in cost.

Section 5

MATERIALS OF CONSTRUCTION

INTRODUCTION

The discussion of construction materials for each energy conversion system is divided into the following parts:

DESIGN PHILOSOPHY

Basic design concepts and design assumptions used by the cycle advocates are considered here. Wherever possible, the component life, estimated by the systems and plant integration team, has been included.

BASE CASE MATERIALS

This information is presented in tabular form for each energy conversion system covered in this section. The materials are those used by the cycle advocates in their design.

MATERIALS REQUIRED FOR EXTREME VARIATIONS

This section discusses material substitutions that will be required for extremes of temperature, stress,or other variables as defined by the parametric variations.

MATERIALS CRITIQUE

This part is an assessment of material problems and potential material problems.

STATUS OF MATERIALS

The status of materials is rated on the scale of 1 to 5, from immature to mature. An immature system is rated 1. Systems of intermediate maturity are rated from 2 through 4. Mature systems are rated 5. A summary of material ratings is presented in Table 5-1. Relatively few material problems are anticipated in mature systems, since they have been designed in detail and are modifications of earlier designs. However in immature systems, those which have no operating experience or detail designs, potential material problems are sure to exist. Systems with intermediate maturity are assigned a number in accordance with their level of maturity.

Table 5-1

SUMMARY OF MATERIALS STATUS

System	Status of Materials (Rating*)	Comments
Simple and regenerative gas turbines		
Air cooled	5	Mature design.
Water cooled	4	Fabrication difficulties in making composite blades. Long term reliability.
Closed-cycle helium gas turbine	4	Data on behavior in "oxydizing" He environment is required.
Supercritical CO_2 Turbine	3	Little real operating experience and lack of data on materials behavior in CO_2.
Advanced Steam Turbine	2	For the base case of 1200 F inlet-1000 F reheat. Fabrication of advanced alloys in the sizes required will be a barrier.
(For 1000 F inlet - 1000 F reheat)	5	The 1000 F inlet-1000 F reheat cases will use state-of-the-art machine.
K liquid metal topping turbine	3	Fabrication difficulties in making superalloy components, such as wheels, in the desired sizes. More tests run in K and Cs are required.
Open-cycle MHD	1	No operating experience or detailed design. Barrier material problems exist in several components.
Closed-cycle inert gas MHD	1	No operating experience or detailed design. Potential severe materials problems in several components.
Liquid metal MHD	1	No operating experience or detailed design. Potential severe material problems but not as bad as with the open-cycle, or inert gas MHD cycles, which operate at higher temperatures.
SPE Fuel Cell	3	0.2 g/ft^2 pt loadings are desired, but not yet achieved.
H_3PO_4 Fuel Cell	5	For cells with lives less than 40,000 hr with approx. 1 g/ft^2.
(For 100,000 hr life cells)	3	For cells with lives of 100,000 hr and loadings of 0.2 g/ft^2.
High-temperature ZrO_2 fuel cell	2	High-performance cell but subject to more materials problems than the SPE or H_3PO_4 cells.

*Rating System
5 = Mature system—current state-of-the-art a mature design—relatively few material problems expected.
2-4 = Intermediate maturity.
1 = Immature system—no operating experience or detailed design - potential barrier materials problems have been identified.

DISCUSSION OF ENERGY CONVERSION SYSTEMS

Open-Cycle Gas Turbines

Design Philosophy. Air and water cooling will be utilized to enable the use of proven gas turbine materials and designs. Specifications consistent with current designs for a fixed bed gasifier have been placed on the maximum carryover of contaminants to minimize hot corrosion and erosion. These specifications are:

	PPM in Combustion Product
Na+K	<0.02
Pb	<0.02
V	<0.01
Ca	<0.04
S	<0.005
Solids	<0.01
All others	<30

The solvent refined coal will contain as much as 0.78% S. This much S, combined with the Na, K, and V in the fuel will cause hot corrosion unless the fuel is treated. The fuel will be washed to remove Na and K. MgO will be added to tie up the V. The estimate for this processing has been included in the economic calculations.

The gas turbine designs are mature and proven. The material selection has been based upon the following factors:

- Creep behavior
- Stress rupture performance
- Resistance to crack initiation and propagation
- Cyclic flaw growth
- High cycle fatigue
- Thermal expansion compatibility
- Thermal conductivity
- Corrosion and oxidation resistance
- Castability
- Fabricability
- Machinability
- Weldability
- Maintainability
- Availability
- Past experience.
- Cost

When cooling is used to limit metal temperatures, and contaminants are limited, the life of an advanced gas turbine can be expected to equal present-day designs. The design life for hot gas components, used in this study, is 50,000 hr. For non-hot gas components, it is 240,000 hr. Baseload operation was used as the groundrule for this study.

Base Case Materials. A breakdown of the base case materials, used on air-cooled gas turbines and water-cooled gas turbines, is presented in Tables 5-2 and 5-3.

Materials Required for Extreme Variations. The use of more cooling for cases with higher operating temperatures was assumed. The cooling is to keep metal temperatures the same as for the base case so no new materials are to be used.

Materials Critique. Because of their mature design and considerable manufacturing experience, gas turbine systems have the fewest materials problems of all the cycles investigated. This is not to say that there are no gas turbine materials problems. However, design approaches to the advanced gas turbines, if successfully implemented, will not require any new materials or material fabrication techniques (except for the composite water-cooled parts or where ceramics are to be used).

There may be problems in fabricating composite blades for the water-cooled turbine. New methods for efficient skin forming and brazing are needed to ensure a good skin fit and to prevent cooling passages from being plugged up.

The long-term reliability of advanced air and water cooling for turbine inlet temperatures greater than 2000 F (1366 K) has not been proven. Air cooling is used extensively in aircraft engines, but their design life and duty cycle is far shorter than that of land-based gas turbines used for power generation. Corrosion (in water-cooled turbines), oxidation, and contamination may destroy or plug up the cooling passages. Long-term reliability will also require new techniques for long time life prediction of stress rupture and fatigue. In some cases, rupture data have to be extrapolated beyond the range where parameters can be validly measured, to design for 100,000-hr lives at high-temperatures. For instance, sigma phase may form after long times but not be predicted from shorter time laboratory tests.

Gas compositions that come from the gasifier, or from cleaned liquid coals, should be such that hot corrosion will not be a problem. It may, however, be economically attractive to trade off gasifier or cleanup costs against more frequent replacement of hot gas components.

The water-cooled turbine is an attractive way to use dirty fuels (residual oils, untreated liquid coal, etc.). The low metal temperatures avoid Na_2SO_4 - K_2SO_4 hot corrosion problems (see the discussion of the open-cycle, MHD cycle, and the primary heat input system). There may be lower temperature oxidation and hot corrosion problems with water-cooled turbine com-

Table 5-2

MATERIALS FOR AIR-COOLED GAS TURBINE

(Base Case 2200 F Inlet Temperature)

Component	Subcomponent	Material	Comment
Compressor	Casing	Nod. Fe	Air cooling is used to reduce the coated vane temperatures for high inlet temperature cases (1600-1800 F for vanes).
	Vanes	403 SS	
	Rotor disc	A4340	Alternates are RA 333 and HS188. Metal temperature is 1550-1750 F (1116-1227 K). Coated blades will be used.
	Blades	403 SS	
Combustor	Cans and flow shield	HAST X	
	Casing	Nod. Fe or 2 1/4 Cr-1 Mo	
	Transition piece	HAST X	
Turbine	Nozzle	X40 or FSX 414	Si-SiC is considered for a 2600 F (1700 K) uncooled nozzle and transition piece.
	Interstage diaphragm	Hn SS	
	Interstage seal	IN-718	
	Shell	Nod. Fe	
	Rotor front Shaft	IN-706	
	Buckets	IN-738 or René 80	
	Discs	IN-706	
	Spacers	IN-706	
	Rear shaft	IN-706	
	Tie bolts	IN-718	
	Cooling fan blades	403 SS	
	Cooling fan disc	A4340	
	Exhaust casing	LC Steel 321 or 347 SS	
	Cooling fan flow guide	304 SS	
	Tie bolts	IN-718	
Air intercooler			
	Tubing	90-10 Cu-Ni	
	Shell, channel, and tube sheet	Steel	

Table 5-3

MATERIALS FOR WATER-COOLED GAS TURBINE

(Base Case 2800 F Inlet Temperature)

Component	Subcomponent	Material	Comments
Compressor	Casing	Nod. Fe	
	Vanes	403 SS	
	Rotor disc	A4340	
	Blades	403 SS	
Combustor	Cans. & Flow shield	HAST X	Alternate - RA 333.
	Casing	Nod. Fe or 2 1/4 Cr-1 Mo	
	Transition piece	HAST X	Cooled, Si-SiC used for ceramic.
Turbine	Nozzle	U500	U500 is used for the core, HAST X for the outer skin and a Cu alloy for the inner skin. In ceramic case, 1st stage nozzle is uncooled and Si-SiC is used.
	Interstage diaphragm	HN SS	
	Interstage seal	IN-718	
	Shell	Nod. Fe	
	Rotor front shaft	IN-706	
	Buckets	A286	A286 is used for the core, HAST X for the outer skin, Cu alloy inner skin.
	Disc	IN-706	
	Spacer	IN-706	
	Rear shaft	IN-706	
	Tie bolts	IN-718	
	Cooling fan blade	403 SS	
	Cooling fan disc	A4340	
	Exhaust casing	LC Steel 321 or 347 SS	
	Cooling fan flow guide	304 SS	
	Tie bolts	IN-718	

ponents. Before water-cooled turbines can be safely run on the fuel in question, it will be necessary to study the influence of fuel impurities on the oxidation and hot corrosion of turbine components at temperatures as low as 900 - 1200 F (756 - 922 K). The thermodynamics and kinetics affecting the formation of deleterious compounds may set the desired degree of cooling and, thus, the operating metal temperatures. It may not be desirable to operate at as low a temperature as possible, since the deleterious compound may not become volatile. In the other extreme, more cooling may be desired to solidify the compound in question and render it relatively innocuous.

Status of Materials. Air-cooled turbines have a maturity rating of 5 (mature). Water-cooled turbines have intermediate maturity (rating of 4). Anticipated fabricating difficulties resulted in the intermediate maturity rating for the water-cooled turbine. Work will be needed to achieve the desired long time reliability. The use of clean fuels may not be either technically or economically realistic. More research into oxidation and hot corrosion and the influence of fuel impurities is required.

Closed-Cycle Helium Gas Turbines

Design Philosophy. Current gas turbine materials with standard designs are employed. Helium cooling for the rotor, shaft, wheels and blades is used to ensure that the metal temperatures are kept to those of conventional turbines. The materials used for this study are the same as were used by the cycle advocate (the General Electric Gas Turbine Division) in a more extensive study of a helium turbine operating in conjunction with a high-temperature gas reactor (HTGR). The questions regarding changes that might be necessitated by not using an HTGR are discussed in the materials critique.

The design life for all the components in the HTGR system was 40 years. It was felt that designing for HTGR helium environments was a more difficult requirement. The reason was that if a 40-year life could be met for HTGR applications, it could be met for a helium turbine, working with other heat sources. The 40-year life is critical for the HTGR application because of the hands-off maintenance requirement. Such a life is not critical for other heat sources and there may in fact be economic advantages of a shorter design life if lower capital costs could be achieved.

Base Case Materials. A breakdown of the materials used for closed-cycle helium gas turbines (base case) is presented in Table 5-4.

Material Changes Required for Extreme Variations. For the 1700 F (1200 K) inlet temperature variation, Mo-TZM will be used for turbine buckets and nozzles.

Table 5-4

MATERIALS FOR CLOSED-CYCLE HELIUM GAS TURBINE

(Base Case 1500 F Inlet Temperature)

Component	Subcomponent	Material	Comments
Compressor	Rotor blades	403 SS	
	Stator blades	403 SS	
	Wheels	NiCrMoV steel	Alternate-MA-250.
	Inlet burst container	CrMo steel	
	Inner casing	Nod. Fe	
	Outer casing	CrMo Steel	
	Shafting	NiCrMoV steel	
	Stub shafting	M-152	
	Distance piece	IN-706	Alternate-CrMoV.
	Appendix	CrMo steel	
	Outer exh. casing	Cast C steel	
	Inner exh. casing	Cast C steel	
	Diffuser	Cast C steel	
	Bearing housing	Cast C steel	
	Bolting	Cast C steel	
Turbine	Buckets	M-21 LC	Alternate-René 100, and IN-738, for 1700 F (1200 K)—Mo-TZM.
	Nozzles	M-21 LC	Alternate-IN-713 LC, for 1700 F (1200 K)—Mo-TZM
	Wheels	IN-706	Alternate-A286
	Spacer discs	IN-706	Alternate-A286
	Inner flow path	HAST X	
	Outer flow path	HAST X	
	Bear housing	Cast C steel	
	Inlet pipe	IN-713 LC	
	Stub shafting	M-152	
	Coupling shaft	NiCrMoV steel	
	Trans. piece	HAST X	Alternate 347 SS
	Piston rings	IN-X-750	
	Inlet piston	HAST X	
	Ring retainer		
	Turbine support ring	CrMoV steel	
	Insulation casings	HAST X	

Table 5-4 (Cont'd)

MATERIALS FOR CLOSED-CYCLE HELIUM GAS TURBINE

(Base Case 1500 F Inlet Temperature)

Component	Subcomponent	Material	Comments
Turbine (Cont'd)	Turbine casing	CrMo steel	
	Labyrinth seal stator rings	Invar	
	Exhaust outer casing	CrMo steel	
	Exhaust pipe	IN-718	
	Exhaust casing	CrMo steel	
	Exhaust seal	CrMo steel	
	Exhaust end well	CrMoV steel	
Pressurizing Gas turbine (Cases 4-8, 12, 24,25,31-33)			Conventional Gas Turbine.* In case 8, where a PFB is used, exposure to coal combustion products may cause severe hot erosion problems. Coated IN-738 and FSX 414 are envisioned for the blades and vanes respectively, for the pressurizing gas turbine.
Helium regenerator	Tubes	304 SS	
	Shells	C steel with 1/4% Mo (Cases 1, 15,16, 40) 2 1/4% Cr - 1% Mo (Case 28)	
Helium pre-cooler/inter-cooler	Tubes	90-10 Cu-Ni	
	Shells, channel & tube sheets	Steel	

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*It was assumed, for purposes of the study, that the gas entering the gas turbine would be cleaned to a purity which would permit employment of conventional gas turbines.

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Cooling may be required to lower the metal temperatures to 1500 F (1089 K) for this high temperature case.

Materials Critique. Almost all consideration of materials serving at high temperatures in He turbines has come from studies performed in conjunction with HTGRs. In reactors, impurities in the He gas are equilibrated with hot carbon from the reactor. This produces a He environment, containing mostly H_2, and CO with traces of H_2O and CO_2 (ref. 1). The H_2/H_2O and CO/CO_2 ratios are set by the graphite core temperature and are expected to be in the range of 25/1 to 250/1. The equivalent oxygen potential for such ratios is low, but not low enough to prevent oxidation of such elements as Al, Ti, Si, Mn and Cr. (Fe, Ni, Co and Cu will not be oxidized.) This low oxygen potential causes intergranular oxidation of the species that can be oxidized. No protective layer of Al_2O_3, or Cr_2O_3 is formed. Rather, there is embrittlement of the localized oxide zone at the grain boundary where Al and Cr segregation may exist. The formation of the local oxide is accompanied by considerable microstructural changes. Adjacent to the oxide is a zone that is denuded in the oxidized species. Below this, a zone of carbide precipitation is often observed. Experiments with HTGR-type He (with the appropriate H_2/H_2O and CO/CO_2 ratios) have given rise to several suggestions for appropriate alloys to be used in such environments (refs. 1 and 2). For instance, high Al and Ti contents are very bad.

The preceding brief discussion of the HTGR behavior has been included so that a contrast can be made to a He turbine operating without a reactor heat source. With no graphite core, the H_2/H_2O and CO/CO_2 ratios in the He turbine will not be the same as that observed in the HTGR turbine. Instead of being reducing (high H_2/H_2O and CO/CO_2 ratios), air leaks are apt to make the He turbine more oxidizing.

It is quite likely that the oxygen potential will be greater than that expected in a He HTGR turbine. For $H_2/H_2O = 250 = CO/CO_2$, the PO_2 can easily be shown from data of the JANAF Tables (ref. 3) to be about 10^{-25} atm (10^{-23} MN/m^2). (For ratios of 25, it is 10^{-23} atm.) It is unlikely that such low oxygen partial pressures can be maintained without the action of the C in the reactor. Only a partial pressure of 10^{-16} atmospheres, or greater, is required to allow NiO to form at 1340 F (1000 K). Only PO_2 greater than 10^{-11} atm (10^{-9} MN/m^2) is required for Fe_2O_3 formation. (Most other constitutents in the alloys require smaller partial pressures.) If the oxygen partial pressure in a He turbine is greater than, say, 10^{-10} atm (10^{-8} MN/m^2), protective oxide layers could be formed. If this is the case, experience with conventional gas turbines will govern the materials choice. The advantage here is that the rate of oxidation will be lower than in combustion turbines, where the oxygen partial pressure is much higher. A study of a turbine, operating in conjunction with an HTGR, was used as the basis for the materials selection for the He turbine. The actual gas composition may make this appear to be an improper approach. However, depending upon the actual oxygen potential, the selection may be conservative.

The experience with conventional gas turbines, or with HTGR materials research, indicates that materials should not be a barrier problem for a He turbine. This will, however, not eliminate the need for considerable materials research.

If the He of the turbine is reducing (as in an HTGR turbine), then the absence of protective oxides will also cause friction, wear, and galling problems. (The oxide is a lubricant.)

The M21 LC alloy proposed for turbine components has no Ta and Co (elements with a very long half-life). Therefore, it is ideal for application in a He turbine operated in conjunction with a high-temperature gas reactor. It is also a reasonable material for a He turbine which does not operate with a gas reactor. It is castable, and preliminary evidence shows that it has acceptable mechanical properties. It is backed up by more conventional alloys, which could easily be used. These may be preferred (because of a larger data base) where Ta and Co are permitted. The final choice will depend upon more design and materials data.

Mo (proposed for variations where the inlet temperature is 1700 F [1200 K]) is a good choice for high-temperature applications in HTGR type He turbines (i.e., in reducing conditions). This is because the equilibrium partial pressure for MoO_3 formation is about 10^{-18} at 1340 F (1000 K). (This is determined from the JANAF [ref. 3] Tables.) At 1000 K, PO_2 for $CO/CO_2 = 250 = H_2/H_2O$ is only about 10^{-25} atm (10^{-23} MN/m^2). (For ratios of 25, it is 10^{-23} atm.) Mo will not oxidize in gas reactor He. In the "oxidizing He" that may be present in a closed-cycle He turbine run without a gas reactor, the poor oxidation resistance of Mo may prevent its use. MoO_3 is volatile and nonprotective. CO additions (to drive the partial pressure of the oxygen down) or the use of other getters may be required. Coating the Mo is another possible approach.

Status of Materials. The closed-cycle helium gas turbine has an intermediate maturity rating of 4. More operating data are required for a mature rating. Questions concerning the influence of "oxidizing He" must be answered.

SUPERCRITICAL CO_2 CYCLE TURBINE

Design Philosophy. Current gas turbine materials are used with this different working fluid. The design life of 30 years is based upon maintaining the stress level below that required by a Larsen-Miller stress rupture analysis. For this Task I Study, no special allowance was made for the CO_2 atmosphere or for fatigue. To consider fatigue, more detailed design and operating information is required than it was possible to obtain in Task I.

Base Case Materials. A breakdown of the base case materials for the supercritical CO_2 cycle turbine is presented in Table 5-5.

Material Changes Required for Extreme Variations. HA188 is the proposed (by the cycle advocate) replacement for IN-718 for Case 9 (1600 F [1045 K] turbine inlet temperature). The 10,000 hour rupture life for HA188, at 1600 F (1144 K), is only about 7000 psi (48 MN/m^2). Hence, it is quite possible that this alloy would not be acceptable for turbine blades, that is, where the stress for the 1350 F (1006 K) inlet temperature was calculated as 6,700 psi (46 MN/m^2). As an alternate, cooling could be used to reduce the metal temperature. However, this would entail a loss of efficiency which was not taken into account in the thermodynamic analysis.

Materials Critique. The data gathered for the use of CO_2 in nuclear reactors provide a base from which to assess the potential problems which may arise in CO_2 turbines. These data (refs. 4 and 5) do not point to any major problem associated with the use of CO_2. With very low CO partial pressures, the behavior in CO_2 may be much like that exhibited in air. (Low CO partial pressures give rise to high oxygen partial pressures.) So long as $PCO_2 >> PCO$, oxidizing conditions can be maintained and the influence of PCO will not be large. (See the discussion of the He turbine for the case where $PCO >> PCO_2$.)

J. E. Antell, et al. (ref. 5) found little difference between Ni base superalloys, tested in pure CO_2, and CO_2 containing 5-10% CO. The exact behavior depended upon the alloy. For a high Cr alloy, such as HAST X, there was no effect. For lower Cr Nimonic 75, some influence was noted. The oxidation rate for both alloys was similar to that observed in air. Oxidation in CO_2 does, however, present one complication not found with air oxidation. The oxidation is accompanied by the reduction of CO_2. There is the possibility of carburization via the reactions of $2\ M\ (\text{metal}) + CO_2 \rightarrow 2\ MO + C$ or $M + CO_2 \rightarrow MO + CO$ and $M + CO \rightarrow MO + C$. The combination of oxidation and carburization is particularly destructive. Fortunately, the susceptibility to oxidation and carburization varies from alloy to alloy and some alloys are quite resistant. This resistance, or lack of it, can be traced in part to compositional factors (ref. 6). More test data may have to be generated for the specific alloys of interest to determine their behavior in CO_2 and develop new compositions if necessary.

The preceding phenomena occur at elevated temperatures. It is also possible to experience low-temperature failures due to CO_2 (and CO) stress corrosion. M. Tanemiera et al. (ref. 7) found that $CO + CO_2 + H_2O$ mixtures produced transgranular cracking in high strength steels. These tests were performed at 20-40 C (266-277 K). This cracking is probably a type of stress corrosion cracking and should not affect components at elevated temperatures. The phenomenon is also limited to low Cr, high-strength steel. It can be eliminated by going to higher Cr con-

Table 5-5

MATERIALS FOR SUPERCRITICAL CO_2 CYCLE TURBINE

(Base Case 1350 F Inlet Temperature)

Component	Subcomponent	Material	Comments
Compressor	Pump	410 SS	Max. temp is 252 F (396 K). Stress at max. temp. is 3500 psi (24 MN/m^2). Max. stress is 38,000 psi (262 MN/m^2) at 155 F (342 K). The low temperatures should prevent materials problems.
Power turbine			
	Nozzle	IN-718 or Inco 617	T = 1350 F (1005 K), σ very low.
	Blades	IN-718 or Inco 617	T = 1310 F (983 K), $\sigma \approx$ 6,700 psi (46 MN/m^2).
	Wheels	IN-718 or Inco 617	Max. T = 1300 F (978 K), σ at max. T = 2,000 psi. Max. σ = 16,000 psi (110 MN/m^2). T at Max. σ = 900 F (755 K).
Regenerator		316 SS	T = 1088 F (860 K). $\sigma \approx$ 12,000 psi (83 MN/m^2).
(Pressurizing gas turbine) (Cases 5-10)			In Case 5, where a PFB is used, exposure to coal combustion products may cause severe hot corrosion problems.
High-temp. regenerator		321 SS Inco 800	(Cases 1, 2, 4-11) (Case 3)
Low-temp. regenerator		LC Steel	
Precoolers/Intercoolers	Tubing, shells, channels, tube sheets	90-10 Cu-Ni	

tents (greater than 9%) or to austenitic stainless steel. This phenomenon has been mentioned to point out the possibility of failure in low-temperature components.

The HA188 may not have enough strength at 1600 F (1045 K) and Mo alloys (such as Mo-TZM) may have to be used. Should this be the case, Mo oxidation in CO_2 may be a limiting problem. CO additions may be required to lower the oxygen partial pressure. Coatings may offer another way to get around this problem.

Status of Materials. The supercritical CO_2 cycle turbine has an intermediate maturity rating of 3. Materials problems should not be a barrier. Component design and fabrication will be conventional. However, lack of operating experience prevents a mature status rating. There is a need for much more test data (especially with respect to operation in CO_2) before we can accurately design a system to meet the desired life. It may be more realistic to design hot gas components for less than 30 years and take the replacement penalty. This problem should be looked at more closely.

ADVANCED STEAM CYCLE TURBINE

Design Philosophy. Going to a 1200 F (922 K) turbine inlet temperatures will require using superalloys and advanced 12Cr and low alloy steels. The design basis with these materials was a 30-year steady operation life. The exact operation procedures must be brought into the picture to set a more accurate life estimate. The number of shutdowns and the rate of shutdown and restarts are extremely important. The extensive calculations, for assessment of such factors on life, have not been made as part of Task I of the study. To some extent, economic tradeoffs must be considered. For instance, the expected life must be trade off against the desired rate of startup.

Base Case Materials. A breakdown of the base case materials for the advanced steam cycle turbine is presented in Table 5-6.

Materials Changes Required for Extreme Parametric Variations. The most severe variation is in Cases 7 and 13. In these cases, a 1400 F (1033 K) reheat turbine must be designed. Fortunately, the pressures are lower in the reheat turbine than in the high-pressure turbine. Hence, the stresses are lower. Nonetheless, superalloys will have to be used in place of the 12Cr and low-alloy steels used for the base case. The large size of the reheat turbine will cause severe fabrication problems. No alternate material has been proposed. Alloys, such as Astroloy (from which 40-in. diameter wheels were fabricated for the American SST engine), may be used.

A286 is proposed for the reheat turbine and diaphragm for the 1200 F (922 K) reheat case. Since the reheat turbine is even larger than the high-pressure turbine, fabrication difficulties may even be greater.

Table 5-6

MATERIALS FOR ADVANCED STEAM CYCLE TURBINE

(Base Case 1200 F Inlet Temperature and 1000 F Reheat Temperature)

Component	Subcomponent	Material	Comments
Main steam valves	Valve bodies	Incoloy 102	T = 1200 F (922 K).
	Valve stems	Incoloy 901	A valve casting, 4-5 times heavier than present castings, would be required. Hence, a large development effort would be necessary to develop a casting procedure. Oxidation of Incoloy 901 could lead to problems of valve sticking.
Main steam piping		Incoloy 102	T = 1200 F (922 K). Fabrication difficulties require a larger extrusion press than is currently used.
Turbine	Outer shell	Incoloy 102 or 12Cr alloy, alloy steel	T = 1100 F (877 K). Casting development required. The outer shells are approx. 18 ft long and 10 ft wide.
	Inner shell	Incoloy 102 bolts-M252	T = 1200 F (922 K). Casting development required.
	High-pressure rotor	Incoloy 901 or A286	Vacuum arc melting is required for alloys. Capacity is not available to make ingots. Sufficient forging capability to break up the cast structure may not be available. (Press. needed is estimated to be 4 times larger than available.) Rotors are approximately 25 ft long- 40-in. diameter. A286 could only be used if cooled.
	Turbine buckets	Incoloy 901 or M252	
	High-pressure turbine diaphragm	316 SS or Incoloy 102	T = 1200 F (922 K). Materials chosen for expansion compatibility with rotor materials.
Reheat section	Valves	12Cr alloy or low alloy steel	T = 1000 F (811 K).
	Reheat turbine	12 Cr alloy or low alloy steel	T = 1000 F (811 K). Rotor has a larger diameter than high-pressure rotor. Approximately 45" diameter vs about 40" for the high-pressure rotor.
	Reheat diaphragm	12 Cr alloy or low alloy steel	T = 1000 F (811 K). Chosen for expansion compatibility.
Pressurizing gas turbine (Cases 21-27)			Conventional gas turbine. For Cases 24-27, where PFBs are used, exposure to coal combustion products may cause severe hot corrosion problems.

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Materials Critique. The primary problem, building a 1200 F (922 K) inlet temperature steam turbine, will be fabricating large components out of Ni base superalloys and advanced ferritic steels. Casting development and an enlarged forging capability will be required.

Present rotors are forged from 250-400 ton ingots. These ingots are poured from more than one heat. The ingot is then forged to break up the cast structure and make the rotor shape. Present-day superalloy ingots are only about 50 tons in size. The production of 250 to 400 ton ingots is far beyond current practice. The production of such a large ingot from multiple heat is complicated by segregation effects and the careful compositional control that must be maintained on superalloys to prevent the formation of undesirable phases, such as [illegible].

Techniques to insure homogeneity and compositional control will have to be developed as well as the facility in which to make 250 to 400 ton ingots. The fabrication of such large superalloy ingots is unique to the production of superalloy rotors; hence a huge development and capital cost would have to be tacked onto the cycle. Such a facility would probably not be completely utilized all year and this would further increase the rotor costs. Even if such an ingot is prepared, the problem still remains as to how to forge it. The desired high-temperature strength of superalloys will raise the necessary forging loads by a factor of 4 or 5. A special forging facility will have to be used. This cost will be borne by the advanced steam turbine cycle. No costs associated with these facilities have been included in the cost estimates.

Oxidation and high-temperature steam corrosion at 1200 F (922 K) will cause problems, especially in close fitting parts such as valves.

The 1000 F (811 K) inlet 1000 F (811 K) reheat turbines are state-of-the-art and will require no development. The 1200 F (922 K) reheat turbine is beyond current practice and would necessitate the fabrication of very large A286 rotors. The 1400 F (1031 K) reheat case is completely beyond current practice.

Status of Materials. The advanced steam cycle turbine has an intermediate maturity rating of 2 for 1200 F (922 K) inlet temperature. The turbine has a mature rating of 5 for the 1000 F (811 K) inlet and 1000 F (811 K) reheat cases.

The design procedures and materials data are quite mature. The intermediate maturity rating for the base case was given because of fabrication difficulties which must be overcome. These are barrier problems, but they are not insoluble if strong programs are mounted in the areas of casting and forging technology. A large steam turbine, with a 1200 F (922 K) inlet temperature, utilizing superalloys, may be feasible but cannot be built today.

The cost of constructing such turbines will be very high.

LIQUID METAL TOPPING CYCLE TURBINE

Design Philosophy. Standard gas turbine materials are used in a relatively benign environment (K containing approximately 20 ppm O_2). The design is for a 40-year life at 80 percent utilization. The materials and stresses were chosen for a 32-year (80 percent of 40 years), 0.2 percent creep life, based upon creep parameter data. The K environment was not considered in estimating the life. (As discussed later, in the Critique, this may not be a serious limitation.) To evaluate fatigue properly, which was not considered, would require more detailed design and operating information than could be supplied in Task I.

Base Case Materials. A breakdown of the materials for the liquid metal topping cycle turbine (base case) is presented in Table 5-7.

Materials Changes Required by Extreme Parametric Variations for 1700 F (1200 K) Inlet Temperature. A 1700 F (1200 K) turbine inlet temperature will require a HA188 casing, and turbine wheels and blades made of Mo-TZM or Mo-TZC. The low stresses at the nozzles may permit the use of HA188 or X-40.

Small size Mo-TZM wheels have been made and tested in potassium. However, a scale-up is required for the sizes required for an actual turbine (40 in. [1.03 m] at least for a potassium turbine). The proposed operating procedure calls for starting the turbine after preheating to above the Mo ductile brittle transition temperature (500-600 F [533-589 K]). This would prevent the imposition of operating stresses on a brittle material. Thermal and residual stresses may still be present below 500 F to cause problems.

The gettering action of K (potassium) will prevent Mo oxidation. But Cs may not be a good enough getter to prevent Mo oxidation. At 1700 F (1200 K), the equilibrium partial pressures of oxygen and MoO_3, K_2O, and Cs_2O are 10^{-13}, 10^{-16}, and 10^{-8} atm, respectively (10^{-11}, 10^{-14}, and 10^{-6} MN/m^2). For saturation of Cs by O_2 the Cs would not prevent MoO_3 formation. With less O_2, the oxygen activity will be less than 10^{-8} atm. Thus it is possible to prevent MoO_3 formation. Even if the Mo is prevented from oxidizing, the Ti and Zr will be oxidized, changing the alloy's properties.

Materials Critique. Liquid metals can interact with materials in several ways. Many metals, such as Ni, will dissolve in liquid metals. Therefore, allowance must be made for this dissolution. The mass transfer of elements (due to dissolution and precipitation) from hot to cold areas can cause problems because of the resulting change in material characteristics. Carbon transport is particularly important since the carbon content plays a large

Table 5-7

MATERIALS FOR LIQUID METAL TOPPING CYCLE TURBINE

(Base Case 1400 F Inlet Temperature)

Component	Subcomponent	Material	Comments
1st Stage turbine	Nozzle	Inco 617, L605, or HA188	T = 1400 F (1033 K), very low stress level. Fabricability, welding, brazing, machining, and forming are the most critical considerations. Bending stresses are low. Low-cycle fatigue may result from thermal stresses.
	Blades	René 77	T = 1335 F (997 K): $\sigma \approx$ 27600 psi (190 MN/m^2). Selected for 0.2% creep strength and prior satisfactory performance in K. Can be wrought or forged.
	Wheel	Astroloy	T = 1335 F (997 K), 22000 psi (152 MN/m^2). This disc alloy has most promising 0.2 creep strength capability and was used successfully in prior "K" turbines. Problems in making large discs can be avoided by designs which keep disc size within current state-of-art. Development of more accurate long-term design data is required to optimize the alloy selection.
2nd Stage turbine	Nozzle	Inco 617, L605, HA188, or Incoloy 800	T = 1319 F (988 K), σ is very low. Less expensive lower temperature capability austenitic steel may prove acceptable in a more detailed design.
	Blades	René 77	T = 1264 F (958 K), σ 39200 psi (270 MN/m^2). Selection based on 0.2 percent creep strength and prior successful use in K turbine. Detail design may permit lower temperature alloy but no major change in cost is anticipated.
	Wheel	Astroloy	T = 1264 F (958 K), σ = 27000 psi (186 MN/m^2). (See First Stage Turbine comments.) Estimated allowable 0.2 percent creep stresses vs design stresses indicate this alloy will also be required for this lower temperature stage.
3rd Stage turbine	Nozzle	Incoloy 800, Inco 625 or 18-8 SS	T = 1243 F (940 K), σ is very low. (See prior turbine stage comments.) Lower temperatures could provide opportunity to use Inco 625, or stainless steels, for lower cost.
	Blades	René 77	T = 1195 F (919 K), $\sigma \approx$ 42600 psi (294 MN/m^2). Alternate blade alloys could be used at lower temperatures.

Table 5-7 (Cont'd)

MATERIALS FOR LIQUID METAL TOPPING CYCLE TURBINE

(Base Case 1400 F Inlet Temperature)

Component	Subcomponent	Material	Comments
3rd Stage turbine (Cont'd)	Wheel	René 41	T = 1195 F (919 K). σ = 26700 psi (184 MN/m^2). Disc temperature and stress should permit use of René 41 (or possibly Inco 718/Inco 706).
4th Stage turbine	Nozzle	Inco 625 or austenitic SS	T = 1170 F (905 K), σ is very low. (See prior turbine stage comments.)
	Blades	Re77	T = 1128 F (882 K), σ = 57,900 psi (400 MN/m^2). Higher stresses at lower temperatures require continued use of René 77 in latter stages. This is based upon parametric extrapolation of short time creep and rupture data.
	Wheel	Inco 718 or Inco 706	T = 1128 F (882 K), σ = 31,200 psi (215 MN/m^2) (See prior turbine stage comments.)
	Pressurizing gas turbine (Cases 4-7, 9, 10)		Conventional gas turbine. For Case 9, where a PFB is used, exposure to products of coal combustion may cause severe hot corrosion problems.

role in establishing the mechanical properties. This problem may be encountered in the K boiler and condenser, where liquid K is present. Most of the recent studies of dissolution (or corrosion) in liquid metals have been performed in Na, in conjunction with the Liquid Metal Fast Breeder Reactor (LMFBR) program. Fortunately, temperatures as high as 1400 F (which might be an extreme clad temperature) have been studied (ref. 8). In general, there is no marked degradation at 1400 F (1033 K) compared with that observed at 1200 F (922 K). Unfortunately, there is relatively little data for corrosion in K or Cs, but it is expected that the behavior should be similar to that observed in Na. Of even more concern than the lack of K or Cs data, is the lack of very much data for temperatures as high as 1700 F (1200 K). Corrosion data, obtained at temperatures as high as 1840 F (1278 K), show no mass transfer in 1300 hours. More data will be needed to substantiate these preliminary results.

The 20 ppm oxygen present in the K is also a potential source of problems. The equilibrium partial pressure of O_2 for K_2O formation is on the order of 10^{-23} atm (10^{-21}MN/m^2) at 1340 F (1000 K) (from the JANAF [ref. 3] Tables). The 20 ppm oxygen present in the potassium is probably below the solubility limit at 1340 F (1000 K). Therefore, the oxygen activity will be even lower than that corresponding to 10^{-23} atmospheres. (For instance, see ref. 8 for calculations of oxygen in Na.) The K should be reducing for elements such as Ni, but may be oxidizing for elements such as Al and Ti. (See the discussion of the He turbine.) Deoxidation of the bulk alloy can cause friction and wear problems. Local oxidation, at gain boundaries, can be a potential cause of failure. Oxygen also influences the rate of dissolution in liquid metals. The greater the dissolution, the greater the O_2 level.

From the aforementioned problems, one should not assume that the operation in liquid metals is extremely difficult. The environmental interaction with rupture and fatigue process appears to be limited. For instance, James and Knecht (ref. 9) have shown that the fatigue crack propagation in type 304 SS (cycled in Na, containing oxygen contents ranging from 2 to 40 ppm) was not a function of the oxygen content and the behavior was similar to that observed in vacuum. This is not surprising given the low oxygen activity. Very low frequency cycling may, however, show the influence of local grain boundary attack should this occur.

There has been considerable experience with the interaction of materials and liquid metals. This has resulted from ten years work with K turbines and from work on the LMFBR program. Over 10,000 hours of testing in K have been achieved. Blade erosion has been controlled. (Mo-TZM is resistant to erosion [ref. 10].) Liquid metal turbines can be operated at temperatures as high as 1700 F (1200 K), provided: the creep resistance in the material is suitable; there is proper control of the oxygen content (using getters) in the liquid metal; instabilities are controlled; and metals with low dissolution rates are used. The materials development for the alkali metal turbines is relatively advanced, but more data are needed.

It is expected that the corrosion and mass transfer in Cs will be similar to that observed in K. Some work in this area will be required.

The fabrication of large diameter discs is a problem, but not an insurmountable one. The disc size is reduced by using Cs. Design modifications have reduced the disc requirement for K turbines to 40 in. (1.03 m). This is achievable with conventionally fabricated Astroloy and was used in the 40-in. (1.03 m) discs for the American SST engines. Powder metallurgy may enable the fabrication of larger discs. However, more development of the powder metallurgy approach will be required, should it be deemed advantageous to go to larger discs.

The large size of K turbine discs must be considered in assessing basic material properties. Ingot segregation, which is carried through to the forging, can result in reductions in the creep and rupture strengths. It is important therefore to use test specimens, made from representative forgings, or castings, when gathering the necessary design data. Specimen size effects must also be considered.

Small Mo-TZM discs have been made and successfully tested in K. However, larger discs will be required for the proposed system. The scale-up to 40 in. (1.03 m) will be very difficult and will require considerable effort (40 in. Mo-TZM wheels will be required for a 1700 F [1200 K] turbine). A redesign, using smaller wheels, may be necessary for this turbine. The long term compatibility and stability of the Mo alloy should be checked, as it should in the other systems which may use Mo alloys at high temperatures for long times. Mo has extremely poor oxidation resistance. However, the oxidation potential in the K is going to be below that for MoO_3 formation. (See the discussion of the He turbine and above paragraphs.) Since Cs is not as good a getter, Mo oxidation may occur if it is used.

The solubility of Mo in K is similar to that of Fe (refs. 11, 12). Hence, there is a chance for Mo dissolution in the hot areas and precipitation in cold regions.

The brittle nature of Mo alloys, at low temperatures, is another problem that must be considered in more detail.

Boiling instabilities can, because of temperature fluctuations, give rise to thermal fatigue failures. Proper flow control can eliminate instability problems.

If leaks in the potassium boiler, or in a potassium condenser-steam generator, should occur, they can present several problems. These should be looked at in detail:

1. The K, or Cs + H_2O, reaction is a potential problem. However, owing to the compressible nature of the metal vapor vs liquid Na, it is believed the reaction will be less dangerous than that which could occur in systems where a liquid phase could react (such as in the LMFBR).

2. Leaks may contaminate the liquid metal and cause material degradation. The influence of such contamination on performance of materials should be considered in any advanced development program in liquid metal turbines. Cleanup systems may be required to prevent any adverse effects of boiler leaks. Leak detection systems, currently being developed for the LMFBR, should be applied to alkali metal turbines. This should be done to prevent any damage by triggering a shutdown or the operation of special cleanup procedures. Contamination occurring when the turbine is opened for inspection and repair will also be an important problem. The liquid metal will be dumped into a holding tank prior to opening up the turbine. However, some will remain because the liquid metal will wet the metal parts. Careful cleaning (which will be expensive) will be required before the turbine can go back into service.

3. Alkali metal leaks into the boiler may cause enhanced hot corrosion over and above the action of the coal gas. Leaks into water (steam) can result in caustic cracking because of the KOH that will be formed.

Status of Materials. The liquid metal topping cycle turbine has an intermediate maturity rating of 3. Basic gas turbine construction techniques are being proposed. A mature rating was not given because of: potential fabrication difficulties, lack of extensive test data or data from operating systems, and the possibility of materials compatability problems with the K and Cs. For the base case, no real barriers are present, but there are fabrication problems to be solved. This is especially so if it is desired to go to larger than 40-in. diameter wheels. For the 1700 F (1200 K) case, there may be more severe problems. A shorter design life for hot gas components may be economically advantageous. A more refined analysis is needed, however, to assess this.

OPEN-CYCLE MHD (MAGNETOHYDRODYNAMIC)

Design Philosophy. This system uses direct firing, with coal slag covering and protecting the electrodes. A 30-year life was assumed. The systems and plant integration team has added a yearly maintenance charge of 20 percent of the capital cost of the generator, diffuser, combustor, slagging boiler and air preheater to account for the expected less than 30-year life. There are, however, insufficient operating data to get an accurate life estimate.

Base Case Materials. A breakdown of the base case materials for the open-cycle MHD is presented in Table 5-8.

Materials Changes Required for Extreme Parametric Variations. The combustor is water cooled and was assumed for all cases. Since cooling is utilized in the channel and it is insulated by the slag, the base case materials was assumed in all cases. The air preheater design and materials vary with temperature. Up to 3100 F (1978 K) MgO and Al_2O_3 were assumed. For Case 26, where T = 3600 F (2255 K), ZrO_2 was assumed to be used in the hottest sections. Here, a graded structure was assumed, with MgO and Al_2O_3 in the cooler sections.

Materials Critique. The materials problems encountered in this system are perhaps the most difficult of all those encountered. The high temperatures and severe nature of the environment will present many challenges. Corrosion and erosion, due to seed and ash impurities in the combustion gas, will cause problems. These will occur even in components where there are no excessive metal (or ceramic) temperatures (such as water-cooled parts or cleanup and seed recovery systems). Materials problems present barriers in the design of the channel, and preheater. They may also present significant problems in the combuster, cleanup and seed recovery systems. (See reference 13 for a review of materials problems.)

The channel life will be determined by the conductor and insulator performance, presently unknown. The channel will almost certainly have to be replaced long before the design life of the plant is reached. Whether or not a given channel life is acceptable will depend upon the difficulty and cost of replacement, including installation and downtime costs. A 30-year life is very unlikely.

The choice of the channel conductor material was made on the basis of experience gathered with coal-fired furnaces and superheaters. In the range of 1050 F (839 K) to 1350 F (1005 K), hot corrosion, due to $Na_3Fe(SO_4)_3$ and $K_3Fe(SO_4)_3$, is a severe problem. Fortunately, the channel operates above this temperature range. At higher temperatures, however, Na_2SO_4 and K_2SO_4 are the corrosive agents. The corrosion is worst when these sulphates are molten. (The minimum melting point for Na_2SO_4-K_2SO_4 mixtures is 1530 F (1105 K); the melting point of pure K_2SO_4 is about 1950 F [1339 K].) However, corrosion is also experienced below, but near, the melting point (ref. 14).

Most of the study of sulfate hot corrosion has been done in Na_2SO_4 or Na and K mixtures. Comparatively little is known about K_2SO_4 hot corrosion. Hence, owing to the high K content (from the seed), a study into the influence of K_2SO_4 is in order. Judging by its high melting point, 1950 F (1339 K), K_2SO_4 should be relatively benign at 1520 F (1100 K). Work will be needed to

Table 5-8

MATERIALS FOR OPEN-CYCLE MHD

(Base Case 2500 F Air Preheat Temperature)

Component	Subcomponent	Material	Comments
Combustor		Carbon steel	Water cooled. Metal temperature reduced to 440 F (500 K) with a stress of approximately 10,000 psi (69 MN/m^2).
Channel	Metal conductor	Inconel (Probably 601)	T = 1520 F (1100 K). The material selection is based primarily upon oxidation and hot corrosion resistance to combustion products, plus 1 percent K seed. The selection of the channel material comes from experience with coal fired boilers. Some short time (500 hr) tests run with tubes exposed to slags of the type found on MHD channels, showed behavior that was comparable to that seen in boiler tubes. See the critique for an expanded discussion of this point. Yielding is allowed in the design since the channel is not a structured member. Thermal cycling will occur and the resultant fatigue cracks can be life limiting if they penetrate to the cooling holes, causing leaks.
	Insulator	Ceramics (Al_2O_3, MgO, or ZrO_2)	The choice will be a function of the degree of ceramic slag reaction and the electrical properties of the ceramic.
	Duct liner	Coal slag	T = 2600 F (1700 K).
Air preheater			
	Matrix	Ceramic bricks (Al_2O_3, MgO, or ZrO_2)	Al_2O_3 and MgO used below 3100 F and ZrO_2 above 3100 F to 3600 F (1978 - 2255 K).
	Container	Carbon steel	T = 350 F (450 K).
Gas cleanup and seed recovery		Metal	T = 350 F (450 K). The exact choice will be governed by the behavior in gas containing seed impurities. The temperature is low enough to consider carbon steel. Corrosion may necessitate the use of stainless steels.
Super conducting magnet	Coils	NbTi in Cu matrix	On site fabrication may be required because of the large size of the magnet.
	Structure	Al-6061	

substantiate this statement. Other agents, such as potassium carbonates, may be damaging and should also be considered.

A nickel base superalloy was chosen for the conductor. If hot corrosion should preclude such a choice, Co base alloys could be used. For increased strength, the Co base alloy could be used as a clad over a higher strength nickel base alloy. The high Cr content Co base alloys, such as FSX 414, give good hot corrosion resistance.

The channel life can also be limited by poor current spread, which can produce hot spots and arcing. Severe hot spots and arcs can vaporize the electrode materials. Less severe hot spots may melt the slag liner and thus cause local hot corrosion damage.

Erosion must also be considered. The proposed channel behavior was determined from experience in static systems. Erosion due to flow at 1500-2000 ft/sec may markedly affect the material behavior.

Extensive channel testing will be required before there is any valid estimate of channel life.

Air preheater materials also represent a barrier. The life of the ceramic preheater matrix bricks is limited by thermal shock, creep, corrosive attack (due to the seed and ash impurities in the combustion gas) erosion, and ash plugging of the holes in the matrix if these are small. The density of the ceramic is an important variable which must be optimized. The greater the density, the better the corrosion resistance but the shorter will be the life due to thermal cracking. No completely satisfactory brick material has been found.

The low-temperature bottom bricks are more susceptible to hot corrosion. (This is due to the presence of liquid slags.) The bricks at higher positions have less of a corrosion problem to overcome but experience higher temperatures. Thus they must have better creep resistance. The different requirements at different heights will give rise to a graded brick structure (different bricks at different temperatures).

More severe problems may be encountered in the preheaters, operating at 3100 F (1977 K) and above (where ZrO_2 must be used in the top sections). Even for the base case, however, corrosion, erosion, thermal shock, and creep may pose severe problems. These will make the attainment of even a 5-year life difficult to achieve.

K from the seed may cause hot corrosion in the gas side of the gas-steam boiler. As has been discussed above, this problem differs somewhat from the usual problem of hot corrosion because of the predominance of K instead of Na or Na + K mixtures.

From a materials standpoint, the water-cooled combustors can probably be relatively standard devices. However, corrosion and erosion problems may cause some difficulties.

Corrosion and erosion buildup of deposits on heat transfer surfaces and fouling will all pose problems in the gas cleanup and seed recovery systems.

Status of Materials. The open-cycle MHD has an immature rating of 1. No long-term prototype testing has been performed. It is, therefore, impossible to get an accurate estimate of component lives or whether the problems discussed above are going to be real barriers. The extent to which channel or air preheater brick are going to be barrier problems is dependent upon the economic replacement interval.

CLOSED-CYCLE INERT GAS MHD

Design Philosophy. The lack of vibrational and rotational energy states in the monatomic inert gas does not allow a coupling of the gas and the electrons. Therefore, the system operates at a lower gas temperature (for the same electron temperature) than a system using diatomic molecules. The lower temperature, and the more inert nature of the working fluid, lessens the materials problem compared to the open-cycle system. However, the influence of the alkali metal seed, which is added to increase the conductivity, must also be considered when assessing the influence of the environment.

The design assumption used was a 30-year operating life. There is, however, no channel test data to support this (true for all the MHD systems). The systems and plant integration team assumed a yearly maintenance charge of 10 percent of the capital cost of the generator and diffuser and a 15 percent charge for the heat exchanger. These charges were taken to account for the expected less than 30-year life.

Base Case Materials. A breakdown of the base case materials for the closed-cycle MHD is presented in Table 5-9.

Material Changes Required for Extreme Variations. The same channel design will be used for all variations (max. temp. = 3900 F [2421 K]). For cases 16 to 22, a ceramic matrix (Al_2O_3) regenerator will be used. (This is similar to the combustion gas heat exchanger, but operates at a lower temperature, with no evacuation.)

Materials Critique. As with all the MHD systems, the channel life may be a problem. The channel life may be limited by a destructive interaction of the W and the combustion gas carryover. For decoupling of the gas and the electron temperature, the diatomic gas molecule carryover has been set at 100 ppm.

Table 5-9

MATERIALS FOR CLOSED-CYCLE INERT GAS MHD

(Base Case 3000 F Channel Inlet Temperature)

Component	Subcomponent	Material	Comments
Heat exchanger		Al_2O_3 (Z_r-O_2 for sections above 3100 F)	The heat exchanger design is based upon the technology developed for steel furnaces. It is expected that, judging from this experience, the topmost 5-10 percent of the brick structure may have to be replaced every five years. A graded structure is planned with ZrO_2 being used only for the high temperatures. One-inch square holes are planned to prevent slagging problems.
MHD Generator	Conductor	Tungsten plated on Ta	T = 2900 F (1866 K). Argon + 0.15 percent Cs atmosphere W chosen for thermionic emission with Cs absorbed layer, and high-temp. strength.
	Insulator	Al_2O_3	See above.
	Diffuser	Al_2O_3 lined steel	T = 1700 F (1200 K). Argon + 0.15 percent Cs environment. See Materials Critique following.
Cs Feed	Injector	Tip will be stainless steel (probably 300 series). A superalloy could be used also but at a high cost.	T ≅ 1000 F (811 K). Liquid Cs environment.
	Pump	Steel	T = 100 F (311 K). Liquid Cs environment. The material is not important because of the low temperature. Cs compatibility is the design limitation.
	Condenser	Steel	T = 250 F (394 K). Argon with liquid Cs film.
Compressor		Standard compressor materials.	T = 772 F (684 K). Argon.
Argon Purifier		Steel, with activated charcoal purification.	T = 100 F (311 K). Argon + combustion gas impurities.
Recuperator for Case #3	- NOT APPLICABLE FOR BASE CASE -	Incoloy 800	T = 1685 F (1191 K). Argon - Argon heat exchange (inside of tubes there will be 0.15 percent Cs present). Stress will be only 1200 psi.
Argon/Water precooler	Tube shells, channels and tube sheets	90-10 Cu-Ni Steel	

(From an analysis of the combustion gas, this is broken down as follows: 80 ppm of N_2, 2 ppm of CO, 9 ppm of CO_2 and 9 ppm of H_2O.) The advocate believes that the CO_2 and H_2O will be dis-associated and tied up in the Cs. Hence, it will be prevented from interacting with the W electrode. The Cs is present as 0.15 percent (1500 ppm), which gives a large reservoir with which to getter the C, O, and H. (The Cs will be cleaned in a cleanup system to prevent a buildup of impurities.)

Cs, which is present as a gas above 1256 F (952 K), is not as good a getter as K, Na, or Li. For instance, at 1340 F (1000 K) the equilibrium oxygen partial pressure for Cs_2O (gas at 1 atm) is about 10^{-12} atm (10^{-10} MN/m^2) compared to 10^{-25} atm for K_2O (ref. 3). At 2900 F (1866 K), Cs_2O is unstable and will decompose to Cs and O_2. (The free energy of formation is positive.) Unfortunately, K_2O is also unstable at this temperature, so neither will getter the oxygen. At 2900 F (1866 K), PO_2 for Na_2O formation is about 10^{-5} atm (10^{-3} MN/m^2). For Li_2O, it is about 10^{-18} atm. The PO_2 for $WO_3{}^2$ formation at 2900 F (1866 K) is about 10^{-8} atm. Therefore, Li_2O would be the choice for the getter at this temperature.

The aforementioned calculations are a bit crude. The complete Cs-W-H_2O-CO_2-CO system must be considered. (H_2O, CO_2, and CO are present, not O_2 as was assumed in the equilibrium calculations cited previously.) If the Cs does not protect the W (which will probably be the case), then W oxidation, due to the H_2O and CO_2, will result (refs. 15 and 16). Also, because of the volatility of the WO_3, no protective oxide layer will be formed. More work on the CS-W-H_2O-CO_2-CO system will be needed to determine whether or not W electrodes will stand up. The addition of Li to the Cs to act as a getter may be one way to reduce the oxygen activity and protect the W.

The particulate carryover from direct coal combustion will be reduced by 90 percent via a cyclone separator. It remains to be seen if this will be good enough and what effect the carryover will have in the heat exchanger or on the channel. Since the gas will contain liquid and solid particles, erosion damage must also be considered as a potential source of trouble.

The advocate does not believe that the use of solvent refined coal (SRC) would present any problems over and above that encountered with the direct coal combustion.

The heat exchanger design was based on the steel industry technology. There, 5-10 percent rebricking is necessary in 5 years. The life of an MHD heat exchanger is unknown. The core size, in the bricks, has been kept large. This resulted in the enlargement of the size of the heat exchanger to minimize plugging.

The mechanical behavior of the channel and Cs feed injector in Cs will have to be investigated. Problems of mass transfer, erosion, and attack due to the environment will be of concern.

Status of Materials. The closed-cycle inert gas MHD has an immature rating of 1. The lack of operating test data makes an accurate estimate of component life impossible. It is also not possible to determine whether or not the above-mentioned problems are going to be barrier problems. The extent to which channel or heat exchanger brick life will be a barrier problem is dependent upon the economic replacement interval.

CLOSED-CYCLE LIQUID METAL MHD

Design Philosophy. The high conductivity of the liquid metal allows for relatively low temperature operation (600-1500 F) (589-1089 K) where there is considerable experience in liquid Na. A design life of 30 years was assumed for preliminary design considerations. A maintenance charge of 10 percent of the generator and diffuser capital costs was assumed.

Base Case Materials. A breakdown of the base case materials for the closed-cycle liquid metal MHD is presented in Table 5-10.

Materials Changes Required for Extreme Variations (1400 F [1033 K] and 1500 F [1089 K] Inlet Temperatures). For Cases 11 and 12, Nb-1% Zr replaces Inconel X 750 and 316 SS in the mixer, nozzle, diffuser, and separator, as follows:

Component	Material	Comments	
		Case 11	Case 12
Mixer	Nb-1% Zr	T=1408 F (1038 K)	T=1508 F (1093 K)
Nozzle	Nb-1% Zr	T=1392 F (1028 K)	T=1492 F (1084 K)
Diffuser	Nb-1% Zr	T=1389 F (1027 K)	T=1488 F (1084 K)
Separator	Nb-1% Zr	T=1388 F (1026 K)	T=1488 F (1084 K)

Materials Critique. The relatively low temperatures and benign environment make for the least anticipated materials problems of all the MHD cycles. Nonetheless, there are potential problem areas.

Compatibility in Na is a potential problem from the standpoint of corrosion as well as mass transfer effects which will influence the material properties.

The erosion of the sprayed W protective layer may be the channel life limiting factor. If the W is eroded, the graphite will be dissolved by the liquid metal and transported to the rest of the system. In addition to the loss of the electrode material, the Na (or Li) transport of the C may cause additional problems. It may change the composition of other components. The Inconel X 750 was chosen for its resistance to this effect. The advocate

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Table 5-10

MATERIALS FOR CLOSED-CYCLE LIQUID METAL MHD

(Base 1300 F Channel Inlet Temperature)

Component	Subcomponent	Material	Comments
Compressor			Same as He compressor for closed-cycle He turbine.
Mixer		Inconel X 750	$T = 1318$ F (988 K). Peak $\sigma = 11,000$ psi. Na + He environment.
Channel	Electrode	Pyrolitic graphite with a 0.1-in. W protective layer. BeCu bus bar is used.	$T = 1280$ F (966 K) in Na + He. W has good corrosion resistance in Na and Li.
	Insulator	Al_2O_3	
Liquid metal-He separator			
	Nozzle	Inconel X 750	$T = 1285$ F (969 K), $\sigma = 8000$ psi, Liquid Na + He.
	Diffuser	316 SS	$T = 1285$ F (969 K), $\sigma = 2,000$ psi.
Separator		316 SS and Inconel X 750	$T = 1285$ F (969 K), $\sigma = 12,000$ psi.
Magnet		Nb-Ti	
He Precooler	Tubing shells, channels, tube sheets	90-10 Cu-Ni Steel	
He Regenerator		Steel	

estimates that the channel life may only be 1 to 2 years. There is, unfortunately, no erosion data to determine the expected channel life.

The presence of both Na and He complicates the analysis of the influence of the environment on the performance of the component materials. Considerable work is needed to determine the influence of the He on material dissolution, mass transport, and corrosion in Na. The two-phase nature of the system will pose an erosion problem which is not encountered in single phase Na systems, as in an LMFBR. This erosion may be a problem, not only for the W layer of the channel, but also for the Na-He separation system.

Na, from carryover into the heat exchanger used with a steam bottoming cycle, will pose the same safety and contamination problems encountered in LMFBR steam generators.

Status of Materials. The closed-cycle liquid metal MHD has an immature rating of 1. The lack of operating experience and test data makes an accurate estimate of component life impossible. It is also not possible to determine whether or not the above-mentioned problems are going to be barrier problems. The extent to which the channel life is going to be a barrier problem depends upon the economic replacement interval.

LOW-TEMPERATURE FUEL CELLS (SOLID POLYMER ELECTROLYTE)

Design Philosophy. These fuel cells use a solid polymer electrolyte (SPE) operating at lower temperatures than cells utilizing H_3PO_4. The electrolyte has the advantage of being non-corrosive (extremely low concentrations of Fl^- ions can be evolved by polymer breakdown, but this should not be too much of a problem). Because the electrolyte is solid, the problem of electrolyte leakage is eliminated, as is the problem of H_2O absorption. Some O_2/H_2 crossover through the electrolyte is possible, but the parasitic loss due to this diffusion is small (estimated at 2%).

The SPE cell operates at a lower temperature than the H_3PO_4 cell and the rate of Pt sintering is thus reduced. Such sintering is a common cause of degradation of cell performance. The lower temperature, however, renders the cell more susceptible to CO poisoning.

A Pt loading of 0.2 g/ft^2 was assumed. This is an extrapolation from current cells, using 1.5 g/ft^2. The assumed life is 100,000 hours. This can be achieved with the present 1.5 g/ft^2 loadings and will probably be possible with 0.2 g/ft^2. However, more work is needed to demonstrate that this is the case.

Base Case Materials. A breakdown of the base case materials for the solid polymer electrolyte fuel cells is presented in Table 5-11.

Table 5-11

MATERIALS FOR SOLID POLYMER ELECTROLYTE FUEL CELLS

(Base Case 170 F Operating Temperature with Air Oxidizer)

Component	Subcomponent	Material	Comment
Electrodes	Catalyst	Pt	34,000 hr lives have been achieved with cells containing 1.5 g/ft^2 operating at 180 F (355 K). There is a cost incentive to go to 0.2 g/ft^2. This lower density Pt loading may reduce sintering, but this has not been proven.
	Anode plate	Ti-Pd	
	Cathode	Ti-Pd	
Electrolyte		Solid polymer (Teflon)	A grafted polymer is under development and was used for the cost estimate. More expensive Nafion is currently used.
Frame		Lexan	
Wet proofing		Porous teflon	
Reformer		Conventional design	

Material Requirement for Extreme Variations (300 F [422 K] operation with O_2 as the oxidizer). The materials requirement is the same as it was for the base case. Successful 800-hour tests run at 300 F (422 K) have shown no degradation in performance. On the basis of these tests, good performance is expected. These cells, however, had more than 1.5 g/ft^2 Pt loadings.

Material Critique. The problems for this cell and for the H_3PO_4 cell are not strictly technical; rather they are economic in origin. These low-temperature fuel cell systems were costed on the basis of 0.2 g/ft^2 of Pt. This assumes a cost of about \$5.5/g of Pt. The Ti-Pd plates of the electrodes are, at \$4/ft^2, a major cost item, as are construction costs. The Pt loading, per se, does not have that dramatic an effect on the fuel cell cost, that is, provided the cost of the Pt is not a function of loading. The widespread use of fuel cells would likely, however, push up the price of Pt. (At present, this is approximately equal to that of gold.) It will thus be important to have as little Pt as possible in the cell. Loadings of 0.2 g/ft^2 are not yet achievable but progress to this level is extrapolated from past experience. Achieving 0.2 g/ft^2 is a barrier which will have to be overcome.

Fuel cell catalysts can be poisoned by S and CO. The S will have to be removed before the gas enters the reformer. Otherwise it will poison the reformer catalyst. The CO will be removed by methanation in the reformer operation.

Status of Materials. An intermediate maturity rating of 3 was given the solid polymer electrolytic fuel cell. Achieving 0.2 g/ft^2 may be a barrier problem.

LOW-TEMPERATURE FUEL CELL (PHOSPHORIC ACID)

Design Philosophy. The use of an H_3PO_4 electrolyte enables a higher operating temperature (375 F [464 K]) than is achievable with SEP cells. The higher operating temperature promotes Pt sintering which can limit life. A 40,000-hour life has been assumed for the design calculation. CO poisoning is not as great a problem with the H_3PO_4 cell, as with the SPE. Therefore, no methanation (to remove CO) penalty was taken. The H_3PO_4 is quite corrosive. This limits the materials that can be used for the construction of the cell. Dilution of this liquid electrolyte is also a potential drawback not encountered with the SPE cell.

A Pt loading of 0.2 g/ft^2 was assumed. This assumption was based upon an extrapolation of present day technology, which uses somewhat less than 1 g/ft^2. Because of Pt sintering, the expected life of this cell may be less than the 40,000 hrs used in cost calculations.

Base Case Materials. A breakdown of the base case material for the phosphoric acid fuel cells is presented in Table 5-12.

Table 5-12

MATERIALS FOR PHOSPHORIC ACID FUEL CELLS

(Base Case 375 F Operating Temperature with Air Oxidizer)

Component	Subcomponent	Material	Comments
Electrode			
	Catalyst	Pt	Expected life lower than for the SPE cell.
	Anode and cathode screws	Carbon fiber	
	Anode and cathode plates	Ti-Pd	Carbon may be used as a less expensive alternate.
Electrolyte		H_3PO_4 in a matrix of boron phosphate and PTFE.	
Frame		Polysulfone	
Reformer		Conventional design	

Materials Critique. Economic constraints will require the minimum Pt loading. Present day technology can make cells using less than 1 g/ft^2. A loading of 0.2 g/ft^2 is desired (see description of SPE cell) and should be attainable.

The higher operating temperature will make the cell life lower for the H_3PO_4 cell than for the SPE cell. Pt can be recovered from the cell, but, as was noted in the discussion of the SPE cell, the Pt cost factor is not dominant in fuel cell costs. A short life will entail a penalty for extra maintenance costs over that considered for the SPE cell. The design life is 40,000 hrs vs 100,000 hrs for the SPE cell. Work may be needed to reach the 40,000-hr life.

Status of Materials. Technically, a mature rating of 5 has been assigned if a Pt loading of 1 g/ft^2 and a relatively short life can be tolerated. The goals of longer fuel cell life and lower Pt loading may be barriers to the large-scale economic use of H_3PO_4 cells. A maturity rating of 3 has been given cells with 100,000-hr life and 0.2 g/ft^2 Pt loadings.

HIGH-TEMPERATURE FUEL CELL

Design Philosophy. Operation at a higher temperature, with a ZrO_2 electrolyte, removes the need for Pt catalysts.

Base Case Materials. A breakdown of base case materials for high-temperature fuel cells is presented in Table 5-13.

Table 5-13

MATERIALS FOR HIGH-TEMPERATURE FUEL CELLS

(Base Case Air Oxidizer with 200 A/ft^2 Current Density)

Component	Subcomponent	Material	Comment
Tube support		ZrO_2	
Electrodes			
	H_2	Ni with ZrO_2	ZrO_2 to prevent sintering.
	Air	Indium oxide doped with Sn	
Electrolyte		Calcia stabilized ZrO_2	
Interconnections		Cobalt chromate	

Materials Critique. Ni sintering can degrade the fuel cell life. Furthermore the air electrode may be a problem. The electrode must be stable in an oxidizing atmosphere with compatible chemical and expansion characteristics. Conducting oxides will be used and possible thermal shock cracking may preclude many shutdowns and startups.

Electrolyte aging reduces conductivity and performance. Also, cell fabrication, particularly with respect to the interconnectors, may be a problem.

Status of Materials. This cell has an intermediate maturity rating of 2. It is a high performance cell that is subject to several life limiting conditions.

SUBSYSTEMS

PRIMARY HEAT INPUT SYSTEMS MATERIALS

These materials are used on conventional furnaces, pressurized furnaces, fluidized beds and indirect heated steam generators.

The primary heat input systems and indirect-heated steam generator parts are designed in accordance with the ASME Power Boiler Code for a 30-year design life. The systems are constructed of more than one material. Different materials are used for different temperature regimes. The exact material selection is based upon the ASME Power Boiler Code design criteria. However, for the uncoded alloys to be used at the highest temperatures, the stress for a 100,000-hr rupture life was used as a design basis. (Suitable safety factors were considered and allowances were made.)

For high-temperature direct-coal firing, there is a significant increase in tube corrosion rate between 1050 F (838 K) and 1350 F (1005 K). Where the cycle working fluid is in this range, the tube wall thickness was increased. This was so that the outside of the tube could be maintained above this range. This necessitated the inclusion of a cost penalty in the estimates because of the greater surface area.

Eight materials were chosen for the materials of construction. These, and the temperature ranges where they are used, are as follows:

Material	Temperature Range
SA-210C	To 1050 F (839 K)
SA-213-T2	To 1050 F (839 K)
SA-213-T22	To 1050 F (839 K)
SA-213-TP 304H	1050 to 1250 F (839-950 K)
SA-213-TP 347H	1050 to 1250 F (839-950 K)
Inconel 601	1150 to 1650 F (894-1172 K)
HAST X	1150 to 1750 F (894-1227 K)
Mo-Re 2	1500 to 2000 F (1089-1366 K)

Of the 3 high-temperature tube materials (Inconel 601, Hastelloy X and Mo-Re 2), only Inconel 601 has been code qualified.

The exact choice for a given temperature would depend upon pressure in the tube. For 1 1/4-in. O.D. tubes, operating at 4000 psig, the following material choices would be made:

Material	Temperature Range
SA-210C	700 F to 850 F (644-728 K)
SA-213-T2	850 F to 950 F (728-783 K)
SA-213-T22	950 F to 1050 F (783-839 K)
SA-213-TP 304H	1050 F to 1250 F (839-950 K)
SA-213-TP 347H	1050 F to 1250 F (839-950 K)
IN-601	1250 F to 1350 F (950-1005 K)
HAST X	1350 F to 1500 F (1005-1089 K)
Mo-Re 2	1500 F to 1650 F (1089-1172 K)

Material Critique. Hot corrosion and erosion of coal-fired boilers are major problems which will require intensive materials research. Depending upon the exact coal composition and alloy to be used, it may be possible to operate fire side temperatures in a region where hot corrosion can be avoided. New alloys may have to be developed, coatings used, or the coal pretreated to remove harmful elements such as S, Na, K and V.

Ultra high-temperature (+1800 F [1255 K]) code qualified material will have to be developed. The ultrahigh-temperature alloy would have to be resistant to the action of coal gas. It may have to be coated. Development of such alloys and coating systems may be a barrier problem in going to the highest temperature systems (1800 F [1255 K]).

Status of Materials. For low and intermediate temperatures, the status is mature (rated 5). There is the problem of hot corrosion which should be looked at in more detail. For high-temperature materials, the status is immature (rated 1).

ORGANIC BOTTOMING CYCLE

The organic bottoming cycle uses a Rankine cycle turbine, operating with an organic working fluid. Fluorinol 85 (85 mole % $CF_3\ CH_2\ OH$–15 mole %H_2O) is to be used with temperatures as high as 611 F (594 K). (Stability has been demonstrated at this temperature.) Freon 22 ($CHClF_2$) can be used up to 430 F (494 K).

The materials of construction are:

Housing casting material	Cast iron CF-8 SS 304 SS
Nozzle block	304 SS
Rotor and shaft	AM355
Heat exchanger tubes	Carbon steel; stainless steel may be used if cooling water corrosion problems are severe.
Boiler, condenser, and regenerator	Carbon steel

This is a low-temperature system, which should produce no problems, providing there is no degradation of the working fluid. Such degradation could develop if air leaked into the turbine and contaminated the organic fluid.

SUMMARY

When considering materials for advanced energy conversion systems, using coal, the following five general material problems must be resolved:

LIFE EXPECTANCY

The design lives of the systems are for 30 years. There are many devices, such as steam turbines, or boilers, with such lives. The problem is that higher operating temperatures are desired. Although materials have been developed for very high-temperature applications, such as those in aircraft engines, the long-term reliability of such materials is open to question. This is especially true where it is necessary to extrapolate beyond the parameter range of rupture or creep data. This is not a very reliable approach. Much more work will be needed to develop techniques for predicting material behavior over very long time periods at very high temperatures.

Cooling can prevent ultrahigh metal temperatures. Air cooling is quite effective for gas turbines. More work will be needed, however, to prove the long-term reliability of cooling where 100,000-hr lives are considered. Corrosion and/or plugging of cooling passages may be life limiting processes.

ENVIRONMENTS

The advanced energy generating systems utilizing coal will require materials to be stressed at high temperatures in a wide variety of environments. The influence of these environments has not been studied in sufficient detail for many of the systems. Of concern, are hot corrosion of boilers and MHD channels, behavior in very high temperature steam, interactions with working fluids that have low oxygen partial pressures, and material behavior in liquid metals. A summary of the environments is given in Table 5-14.

FABRICATION

Several of the advanced energy generating systems will require the fabrication of superalloys into extremely large parts. Such fabrication is presently beyond many present-day practices. Considerable effort in the areas of casting, forging, welding, and powder metallurgy will have to be expanded in order to learn to fabricate the necessary equipment. New alloys may have to be developed to allow for successful fabrication while still maintaining the desired mechanical properties and resistance to environmental factors.

HIGH-TEMPERATURE MATERIALS

There is a need for much more work on materials that can be used where the metal temperature is 1800 F (1255 K) and above.

Table 5-14

OPERATING ENVIRONMENTS

Environment	System
Coal gas containing hot corrosion-producing elements (S, alkali metals)	• Coal-fired boiler tubes
	• Open-cycle MHD
	• Pressuring gas turbines for PFB
Clean coal gas and air or combustion product from cleaned liquid coals	• Simple and regenerative gas turbines
Inert gas (He, Argon,	• Helium gas turbine
	• Inert gas closed-cycle MHD (Argon) (Cs also present)
CO_2	• Supercritical CO_2 turbine
Liquid metal (K, Cs, Na)	• Liquid metal topping turbine
	• Liquid metal closed-cycle MHD (Helium also present)
High temperature steam	• Steam turbine and heat exchangers
H_2, O_2, Air	• Fuel cells
Organic fluid (Fluorinol)	• Organic bottoming cycle

Many large groups of alloys, such as low alloy steels, and ferritic, or austenitic, stainless steels, can be used at 1000 F (811 K) and below. The poor high-temperature strength of low alloy steels does not permit use above 1000 F (1811 K). Austenitic stainless steels are only marginal between 1000 F (1811 K) and 1200 F (922 K). For temperatures of 1000 F (810 K) to 1500 F (1088 K), Ni or Co base superalloys can be used. A few specially modified superalloys may have useful operating temperatures up to about 1800 F (1255 K). However, above 1800 F (1255 K), refractory alloys and ceramics will have to be used. The use of ceramics and some advanced superalloys and refractory alloys, are limited by their brittle nature at even the highest temperatures. Ways of designing for this brittleness and working with brittle materials must be established.

The refractory alloys, which may be ductile at the operating temperature, have high ductile to brittle transition temperatures. These impose design limitations which must be avoided by using such devices as hot starting. The most severe problems to overcome, before refractory metals can be widely used, is their poor oxidation resistance (and resistance to hot corrosion) and fabrication difficulties. The former may be overcome by coating or changing the oxygen potential of the environment where oxidation in inert gases or liquid metals is the problem.

CERAMIC HEAT EXCHANGERS

More work is needed to develop ceramic heat exchanger materials. These will have to be resistant to corrosion and erosion in coal gases as well as resistant to thermal cracking failures.

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